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THE ENGINEERING EDUCATOR

A COMPREHENSIVE, PRACTICAL AND AUTHORITATIVE
GUIDE FOR ALL ENGAGED IN THE ENGINEERING INDUSTRY

Edited by

W. J. KEARTON, M.Eng., A.M.I.Mech.E., A.M.Inst.N.A.

ASSISTED BY A LARGE NUMBER OF SPECIALIST
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VOLUME III

FULLY ILLUSTRATED WITH MANY PHOTOGRAPHS
AND DIAGRAMS

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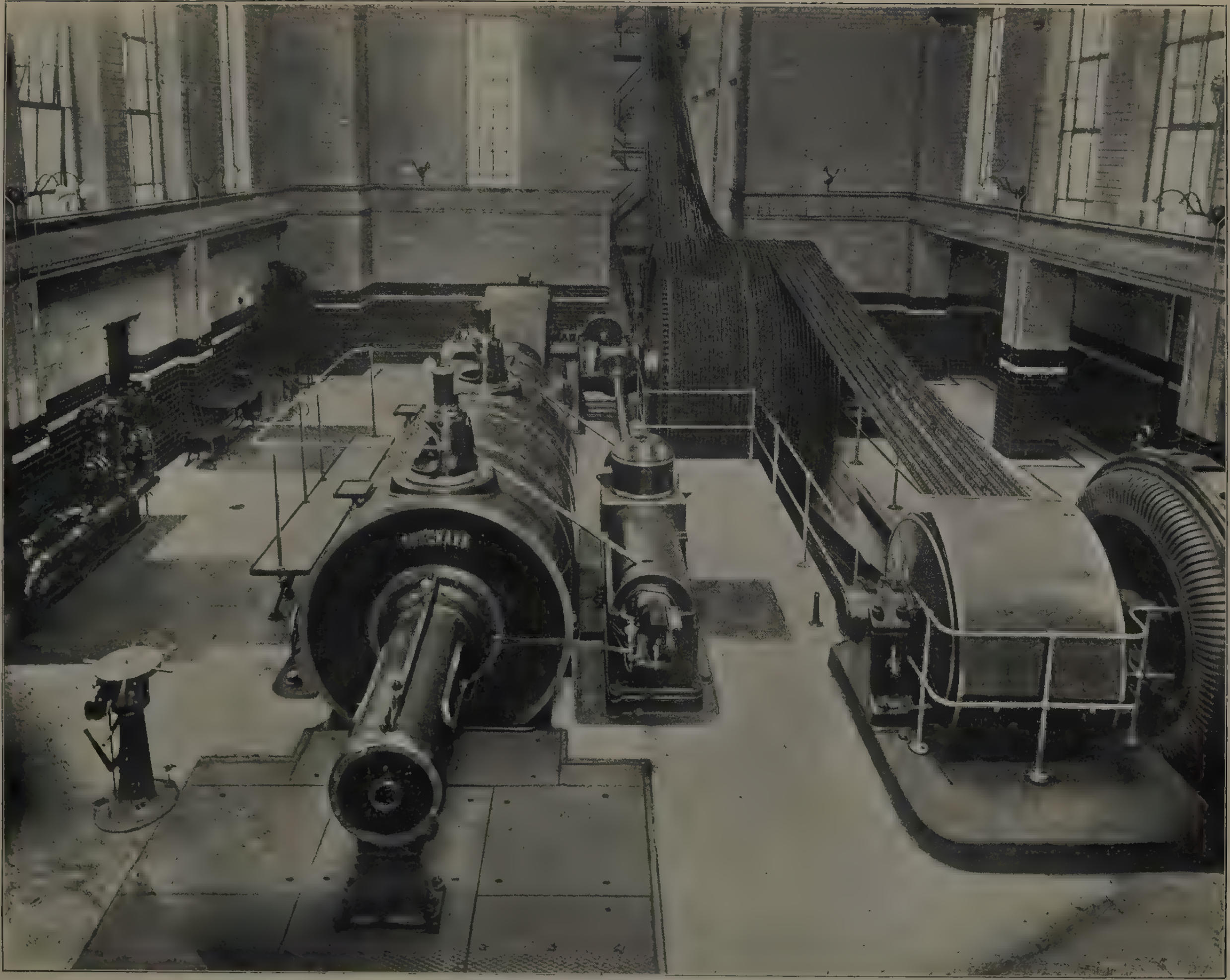

FIG. 30. 1,620 I.H.P. UNAFLOW STEAM-ENGINE

(Messrs. Barlow & Jones, Ltd., Prospect Mill, Bolton)

RECIPROCATING STEAM-ENGINES

By Engineer Lieut.-Commander T. Allen, R.N. (s.r.)

LESSON IX

THE UNAFLOW ENGINE

Principle of Operation. The production of the Unaflow engine on a commercial basis about the year 1909 marked an important event in the history of the steam-engine. The original patent of Leonard J. Todd dates back to 1885, but no general application of the principle appears to have been made during the intervening period. The Unaflow cylinder differs radically from the older types as will be seen from the comparative diagrams, Figs. 26 and 27. Fig. 26 shows, diagrammatically, a section through the ordinary four-valve cylinder, and Fig. 27 is a section through a Unaflow cylinder. It will be noted from the latter diagram, that for each side of the piston, the cycle of the Unaflow cylinder is completed without reversal of flow of the steam; which is admitted and cut-off by the inlet valve, expanded, and exhausted through ports at the centre of the cylinder. Exhaust valves are not necessary, and the direct path of the steam is the basis of the term "Unaflow"; as distinguished from the alternating flow of the ordinary cylinder, in which steam enters and expands during the forward stroke, and is reversed in direction and exhausted during the return stroke.

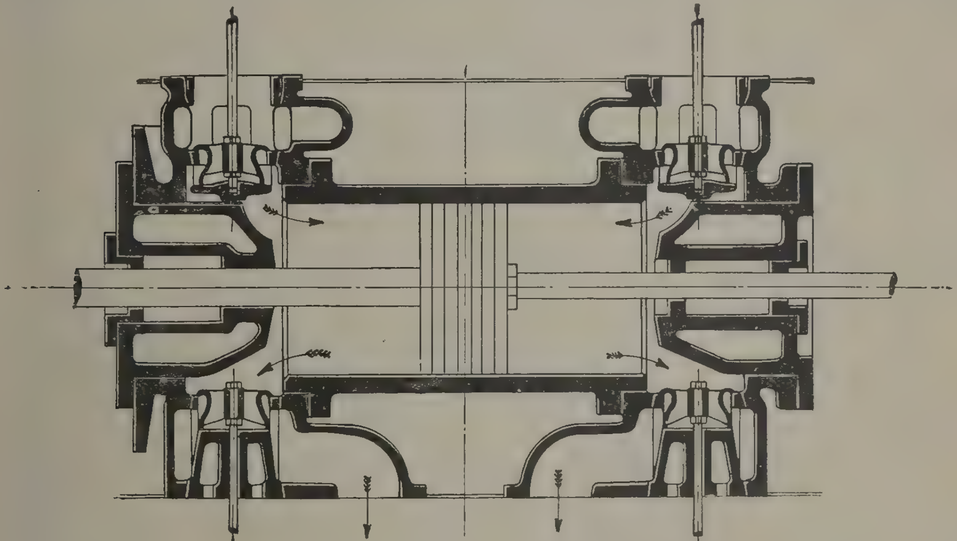

Fig. 26. Diagrammatic Section of Four-valve Cylinder

Cycle of Events. Fig. 28 shows typical indicator diagrams taken from a Unaflow engine cylinder at nearly full-load. For each side of the piston the Unaflow cycle is as follows—

1. *Steam admission* just before the dead centre and cut-off, for normal load, at about 10 per cent of the stroke.

2. *Expansion* after cut-off to about 90 per cent of the stroke, when the exhaust ports are uncovered by the piston, which has a length equal to ·9 of the stroke.

3. *Evacuation* of steam through the exhaust ports from the end of expansion to the beginning of compression which occurs when the piston again covers the ports after traversing about 10 per cent of the return stroke.

4. Compression during the remainder of the return stroke until the admission valve opens to "lead."

Advantages. The absence of exhaust valves permits of reduced clearances, so that a high degree of vacuum may be utilized. Condensation is reduced by the smaller clearance and by heating of the metal surfaces during the compression. Condensation is also greatly reduced, due to the diminished cooling of the cylinder walls and clearance surfaces, by the exhaust steam in its passage from the cylinder. In this, lies the principal explanation of the fact that, after the lapse of nearly a century, the steam-engine has reverted to its simplest form as a single-cylinder machine—notwithstanding the previously accepted view that the highest thermal efficiency was to be obtained by dividing the expansion over a number of cylinders. Other important advantages of the Unaflow engine over the older types are as follows—

(*a*) Improvement of the cycle due to the nearly uniform fall of cylinder temperature between inlet and exhaust. Reduced radiation losses from pipes, cylinders, receivers, etc.

(*b*) Elimination of throttling and increase of power made possible by the absence of exhaust valves and almost complete equalization of cylinder and air pump pressures at the end of the stroke, due to the large area of exhaust ports and the close proximity of the air pump.

(*c*) Good drainage conditions, lower mean cylinder temperatures with higher possible superheats, and reduced cooling due to short exhaust port opening.

(*d*) High mechanical efficiency, low oil consumption, and great simplification of design, due to the reduced number of cylinders, valves, packings, reciprocating parts, etc. Smaller foundations, space, and engine-house requirements.

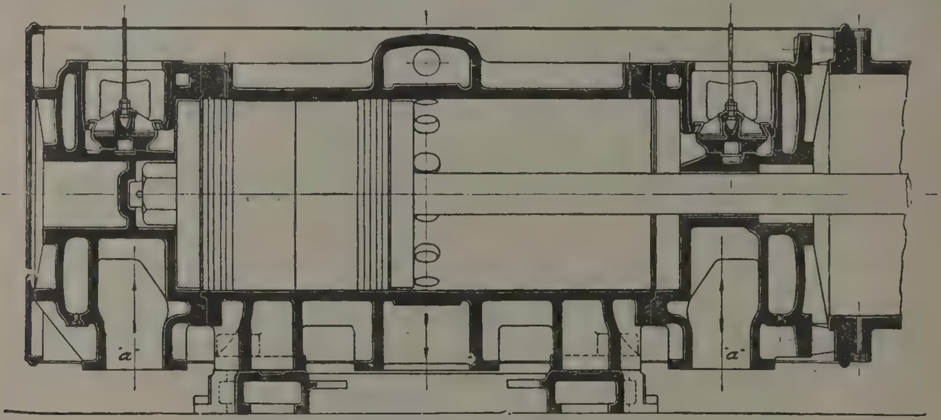

FIG. 27. DIAGRAMMATIC SECTION OF UNAFLOW CYLINDER

General Features. Owing to the high ratio of expansion in a single cylinder, the cut-offs are small and the cylinders large. Although, for reasons of economy the normal load should be developed at about 10 per cent cut-off, the valve gear, under the control of the governor, is designed to give automatic adjustment to smaller or greater cut-offs in accordance with the load. As the range of cut-off is usually governed up to about 25 per cent of the stroke, and an overload of 25 per cent may generally be developed with a cut-off of about 15 per cent, it will be seen that the engine responds readily to varying load conditions.

Owing to the high compression ratio, the pressure at the end of compression with normal vacuum may approximate to that of the initial steam. As the compression ratio is fixed, it is obvious that means must be provided to avoid excessive compression in the event of valve leakage, or undue reduction in vacuum due to failure of the condensing plant or any other cause. Similarly, if during starting and stopping, the vacuum is low or non-existent, or the engine is to run non-condensing, the same necessity arises. In some cases, special automatic valves actuated by the back pressure, and operated from the layshaft, are fitted to by-pass steam to exhaust so as to relieve the pressure—the valves remaining inoperative when the vacuum is good. This arrangement introduces a number of additional working parts, and some of the leading makers employ automatic spring relief valves, together with hand-operated valves for connecting the cylinder, when required, with additional clearance spaces in the covers, so as to relieve excessive compression.

Description. Photographs of Musgrave Unaflow engines are shown in Fig. 29 and in the inset plate facing page 969 (Fig. 30), and from these photographs, the general features of the type will be apparent.

As far as the writer is aware, the engine illustrated in the inset plate (Fig. 30) is the largest normal speed Unaflow engine running in this country. It was built for a normal load of 1,620 i.h.p. when supplied with steam at 170 lb. gauge pressure, superheated to about 600° F. Power is transmitted from a rope fly-wheel on the crankshaft to an alternator and to the card and spinning rooms of a cotton mill—for which duty steam consumption and good governing are points of prime importance. The engine is fitted with a jet condensing plant, incorporating an electrically driven centrifugal water extraction pump, and duplicate steam operated ejector air pumps of the Musgrave "Radojet" type. These air pumps are accommodated on the engine-room floor, near the door shown in the illustration, and the control pillar and electric stop motion are also clearly visible in the photograph.

The engine shown in Fig. 29 is of interest as illustrating the application of Unaflow engines to the direct-driving of large mine ventilation fans. The installation is designed to provide for the increased output and air pressure required from the fan, upon the development of the mine. For this reason, the engine is fitted with a special speed governor, shown in the photograph, which will permit the engine speed to be increased as the workings develop, from about half speed to the full speed of 130 r.p.m. The latter speed is fixed by the speed of the fan, which is direct-coupled to the engine crankshaft by means of a rigid coupling. No fly-wheel is fitted, as the necessary fly-wheel effect is incorporated in the fan rotor. The casing shown near the fan duct covers a light barring rack mounted on the crankshaft between the engine and the fan.

The valves of Unaflow engines are of the drop valve seating, or piston type, driven usually by semi-positive gear through eccentrics and rods from the layshaft. The valves may be lifted by means of cams and rollers, as in the photograph, by oil pressure, or through trip gear; although the latter gear is not often employed owing to the high speed of rotation and small cut-off. The valves are usually returned to their seats by spring dashpots in the valve bonnets, but, occasionally, mechanical closure is effected by closing cams and rollers. The valve gear may, if desired, be arranged for greater cut-offs than 25 per cent, but a higher maximum than this is not required or desirable for high pressure condensing Unaflow engines. Where the gear is applied to back pressure or steam extraction engines greater cut-offs are required, and the gear may, as a rule, be designed to cut-off up to 60 per cent or more of the stroke.

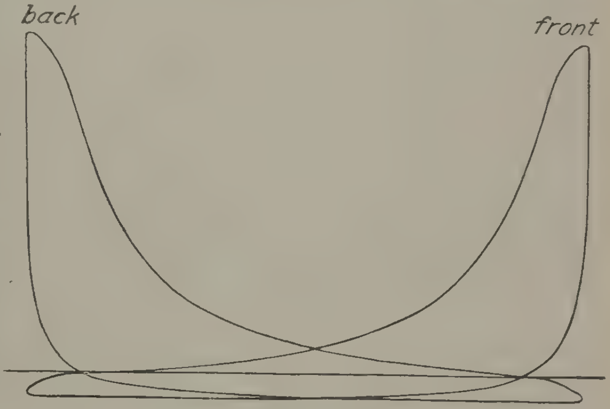

Fig. 28. Typical Indicator Diagrams for Unaflow Cylinder

Governors are of the centrifugal, inertia, or shaft types, driven usually by the layshaft, and arranged to vary the cut-off through regulation of effective cam travel, or point of trip. As all the power is developed in a single cylinder the control of the governor is particularly effective. Hand adjustment devices for regulating the speed, say, ± 5 per cent while the engine is running, are frequently fitted.

Unaflow engines run at speeds between 100 and 200 r.p.m., according to size, the cylinder covers are heated by the passage of steam to the valve chests, taper-boring of the cylinder allows for the greater expansion of the piston and hot ends than of the cold exhaust belt, and the engines are totally enclosed and forced lubricated throughout. Special features of the engines shown in the illustrations are—

Small contact area between cylinder cover and frame bosses, baffling of leakage from front metallic packing, and water-cooling of crosshead race and main bearings, to secure cool frames and bearings, and low oil temperatures.

Flexible admission valve seats to compensate for slight valve distortions due to high superheat or varying loads.

Transmission of governing forces through oil relay to eliminate governor strains, and to afford ample forces for regulating the valve gear.

Flexibility of crosshead and tail rod bearing to allow for temperature changes and any slight inaccuracies of alignment.

Connection of piston-rod to crosshead by adjustable locking nuts, permitting adjustment of cylinder clearances.

The engine frame is normally bolted down to concrete foundations and the cylinder provided with sliding feet, except where flexible oil supports are adopted. The adjusting and anti-friction devices of high-class slow and high-speed engines are retained, similar safety devices may be fitted, and special oils, packings, and cylinder metals are employed to enable the engine to work with high degrees of superheat.

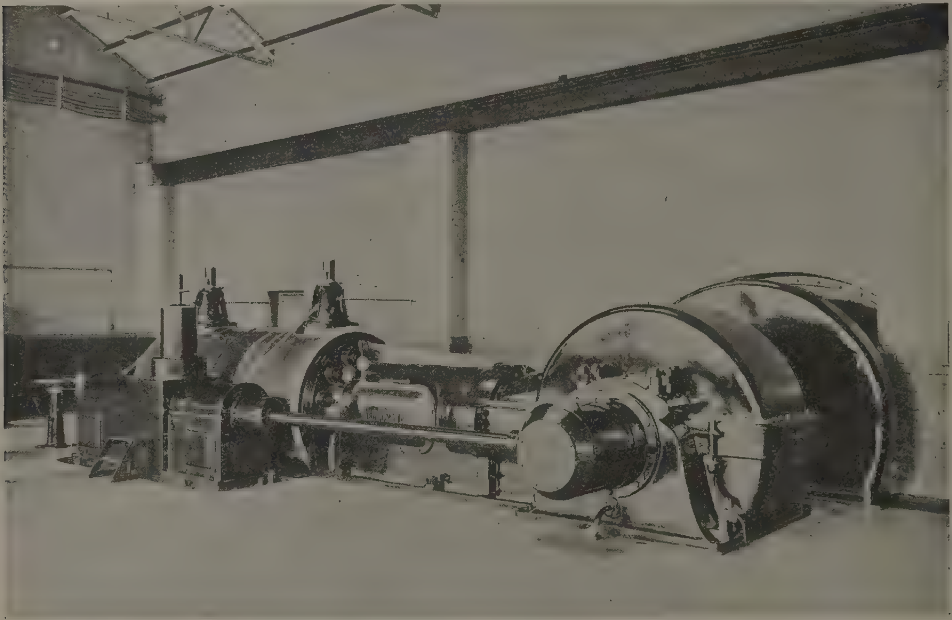

Fig. 29. 820 I.H.P. Musgrave Unaflow Engine Direct-driving Mining Fan

Regularity of Turning. The component parts of Unaflow engines must be somewhat heavier than corresponding parts of compound expansion engines, owing to the higher initial piston loads introduced in single-cylinder engines, by the larger pressure difference between opposite sides of the piston. The massive parts, combined with the greater speed are, however, of indirect advantage, as the high inertia loading tends to equalize the turning effort on the crank pin. With a suitably designed fly-wheel, the cyclical speed fluctuation can be kept within small limits, and the engine is, in every respect, suitable for drives requiring extreme regularity of turning, e.g. alternators, spinning rooms, etc.

Steam Consumption. Wherever possible the Unaflow engine should be worked with superheated steam, as the consumption is greatly affected by the degree of superheat. Steam pressures range, in practice, from about

100-250 lb. per sq. in., and steam temperatures up to about 650° F. The engine will give good results where only low steam pressure is available, and it is equally applicable for the higher pressures. Developments in some designs tend towards the complete oil operation of valves and governing.

A particularly valuable feature of the Unaflow engine is the uniformity of the steam consumption over a wide load range. Between half load and 10 per cent overload the steam consumption per horse-power hour only increases about 5 per cent, and this is of particular advantage in engines running at times on partial load, e.g. generating sets.

Curves showing the steam consumption of a high-class Unaflow engine of about 800 i.h.p. normal load, when supplied with steam at 180 lb. gauge pressure, saturated, and superheated to various total temperatures, are given in Fig. 31. These curves illustrate the remarkable economy of the engine, the uniformity of the steam consumption, and the influence of superheat in reducing the amount of steam used.

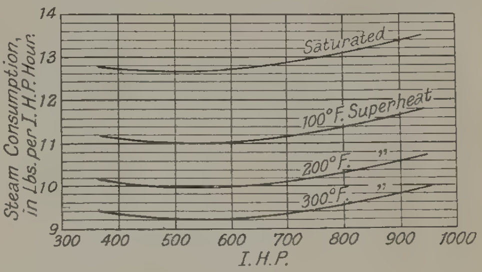

FIG. 31

Curves showing steam, consumption of 800 i.h.p. Unaflow engine. Steam pressure 180 lb. gauge, dry-saturated, and with various degrees of superheat. Vacuum 28.25 in.

OUTLINES OF LOCOMOTIVE ENGINEERING

By A. MORTON BELL, O.B.E., M.I.MECH.E., M.I.LOCO.E.

LESSON III

THE BOILER

LOCOMOTIVE boilers are now almost exclusively constructed of steel, with as few joints as possible. Plates can be procured of such dimensions that the smaller size boilers can have the barrel made in a single piece with but one longitudinal seam. Electric welding enables a still further reduction of joints being obtained. The barrel of the boiler is attached to the outer shell of the fire-box, formed by an outer " wrapper " plate, a " throat " plate, to form the inside end, and a front plate, the outside. The internal fire-box in which the fire is maintained is, in British practice, usually of copper, and this is supported from the outer fire-box by a number of screwed stays, or bolts, which take the stress, or strain, imposed by the pressure existing between. From the fire-box a number of flue tubes pass through the barrel of the boiler and enable the heat generated by the fire to be imparted to the water for the purpose of generating steam. The shape and size of a locomotive boiler is to a large extent dictated by the limited space it must occupy. Rapid combustion of the fuel is essential to secure the necessary generation of steam the engine requires, and to stimulate and obtain this artificial draft is necessary. The exhaust steam from the cylinders is used as a blast at the smoke-box end, inducing the passage of the products of combustion through the tubes, and causing atmospheric air to be drawn into the furnace (or fire), through the ashpan, which is arranged with suitable controlling dampers.

To ensure good combustion at the required speed, considerable reliance has to be placed on the means and methods of firing. Maintenance of a high temperature in the fire-box is essential to good combustion, and it is usual to build in it a fire-brick arch over which the partially consumed gases from the fire must pass before they enter the tubes; it must be remembered that the latter, having water behind them, tend to cool off the gases and restrict further combustion rather than augment it.

There are many designs of fire-box, popular ones in this country at the present time are the "Belpaire," and the direct stayed "round top." The former has a flat top, which secures for it increased heating surface with a very clear steam space, easy to keep clean and free from dirt. The fire-grate is formed of a series of flat bars, usually of cast iron, arranged longitudinally on suitable bearers place at the bottom of the fire-box. This grate has been the subject of considerable modification abroad, due to the low class of coal used and the larger consumption per square foot per hour such fuel calls for; various methods of "rocking" or moving the bars, to break up the fire are almost always employed. Means are also provided for rapidly dropping the large quantities of ash and cinder, which accumulate so quickly on large power units.

On the fire-grate of an ordinary British locomotive, coal can be burned at the rate of from 30 to 200 lb. per sq. ft. per hour.

The total heating surface of a boiler is the number of square feet contributed by the external surface of the boiler tubes plus the area of the fire-box plate exposed to contact with the water. The heating surface of the superheater elements is usually given separately, and as additional. The most valuable part of the total heating surface of a locomotive boiler is that portion represented by the fire-box.

The actual efficiency of a locomotive boiler of good design may be taken as 65 per cent. It is capable of evaporating from 7 to 8 lb. of water per lb. of coal, when good Yorkshire steam coal is used, and its out-turn will rise to 8 to 9 lb. of water per lb. with best Welsh steam coal. About 12 lb. of water per sq. ft. of heating surface per hour is the usual evaporation power of a locomotive boiler, whilst each square foot of grate area may be assumed to account for the evaporation of from 800 to 1,000 lb. of water per hour.

An accepted authority estimates the efficiency of the boiler tube surface as follows: Assuming a fire-box temperature of about 2,000° F., a smoke-box temperature of about 500° F., and a heat transmission from the tubes of 4,500 B.Th.U.'s per lb. of fuel burnt, a theoretical evaporation of 4·7 lb. of water from and at 212° F., would be attained with the firing of 40 lb. of coal per sq. ft. of grate, and a ratio of heating surface to grate of 60 to 1, or, similarly, with a firing rate of 80 lb. per sq. ft. and a ratio of 85 to 1.

The heating surface of a modern locomotive is usually in the ratio of about 70 to 1.

The shell, or "barrel," of the boiler must be large enough to accommodate the number of tubes deemed necessary to give the required heating surface, and also provide a sufficiency of steam space. It is advisable that the tubes should not be too small in internal diameter, or cramped together too closely in their arrangement, otherwise there will be a risk of poor circulation of the water, for the violent ebullition set up between the tubes will tend to keep it away from contact with the heated metal surfaces.

Although it is no longer possible to provide a lofty dome, due to the limitations of height imposed by the loading gauge, a slightly raised and capacious chamber, still called a "dome," is provided, wherein the regulator is located, and the steam for the cylinders collected and taken as far from the surface of the water as possible.

The regulator is now usually of some well known balanced type, with cylindrical valves, as these are so much easier to operate than those of the sliding pattern; further, they are better to maintain and keep tight. As one valve must be slightly larger than the other to permit of their introduction into the regulator head and rest on their correct seatings, this fact is taken advantage of to secure a tendency to keep the regulator shut by the excess of pressure acting on the larger valve.

From the dome all other steam supplies, including the feed injectors, brake pump, steam heat, etc., are usually taken, and, further, the large cover, or manhole, provides a very convenient means of access to the boiler, when required. The dome itself is formed of either pressed steel plate or a built-up detail, whilst the cover is usually in the form of a steel casting.

It will be noted in our illustration (Fig. 3), facing page 921, of a typical British engine, that no dome is provided, but the particular form of the tapering boiler shell tends to secure an equivalent space for dry steam.

The tubes are secured in the tube plates by expanding, beading, and the introduction of protecting steel ferrules. Electric welding has been introduced for this work, and in America, where iron and steel tubes are exclusively used with steel fire-boxes, it is now quite common practice to fix them by this method.

In designing locomotive boilers, provision should be made for the additional call for steam, over and above that made by the engine, for the working of the continuous brakes, either air or vacuum, the heating of the train in the case of engines destined for passenger service, and the various auxiliaries now often operated by steam from the locomotive.

To prevent any excessive pressure being raised in the boiler, safety valves must be provided. These were usually of the so-called Ramsbottom type, wherein two valves of similar size, about 3 to 3½ in. in diameter, were placed side by side on short hollow columns, between which a coiled spring was installed; one end of this was attached to a fixture on the base, whilst the other was connected to a bar crossing the valves, and terminating at one end in a "fish-tail," long enough to enter the engine cab. As the bar, or lever, was pivotted at the point where the spring was attached, between the two valves, lifting it pressed down the valve nearest to it and relieved the more distant one, and vice versa.

Such a device as that described was doubtlessly good whilst pressures were moderate, but it will be evident on a careful consideration that whatever pressure the valves were set to commence to blow off at, that pressure would be increased as the resistance of the spring would be more as the lift of the valve developed; in other words, the controlling influence would not be constant.

To remedy the vagaries of this construction, safety valves of the "pop" type have been designed, and are now generally used. In these the valve is actuated, or caused to rise, immediately the steam reaches the predetermined pressure. The action is simple: the valve, loaded by a suitable spring, is surrounded by an annular chamber on the increased area of which the steam which has lifted the valve can be effective, and add to the force which keeps it raised until such time as the pressure in the boiler falls to the approved maximum, when the valve will instantaneously close. To put the matter very concisely, a "pop" safety valve will lift at the appointed pressure, allow all excess steam to escape, and practically shut at the same figure that it opened at.

Other fitments to the boiler include water gauges, whereby the level of the water in the boiler can be readily seen; these are usually of a "protected" type. The glass column is surrounded by stout plate glass strips, which prevent broken glass flying about should the glass tube burst. Further, the valves which

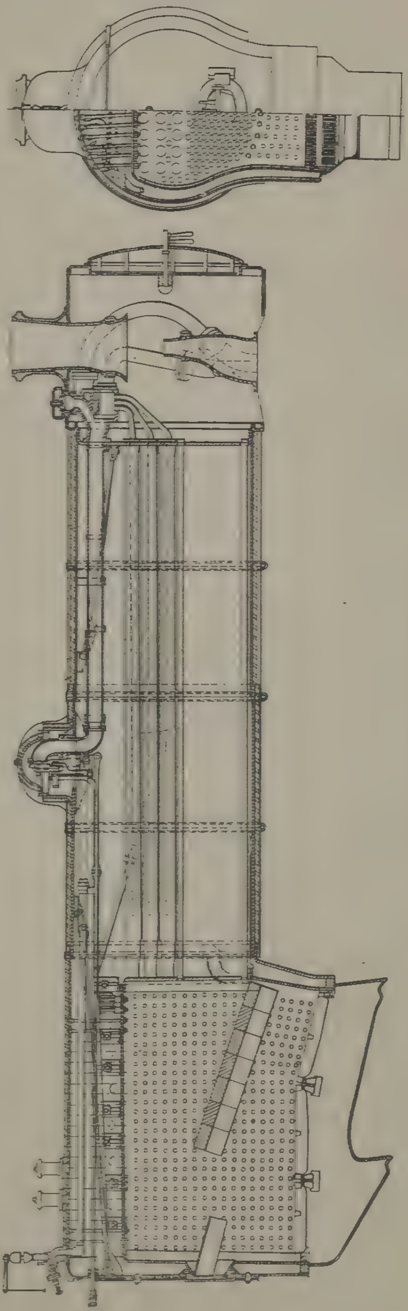

Fig. 11. Section of Boiler with Superheater

control the steam and water passages have automatic arrangements, whereby should a rush of steam or water take place due to fracture of the tube, they will automatically close, and so prevent scalding of the engine-men who may be near by. A refinement consists of markings on the protectors which become magnified when viewed through the water in the tube; there are many different designs.

Among the minor attachments required on locomotive boilers are mud plugs and doors, by which periodical wash-outs and cleaning can be effected. For the holes located above the upper rows of tubes in the smoke-box, and also those at the level of the crown of the inner fire-box, tapered plugs, with fine screw-threads, are generally employed, and it is customary in many instances to provide them with renewable seats, so that if the screw-threads get damaged by the frequent cleaning operations, the seats can be replaced and tight plugs guaranteed. For the apertures along the base of the fire-box, from which considerable quantities of mud and scale may be taken, small hand holes are provided by many designers, which are closed by plates held by a bolt to a bridge piece outside.

Locomotives attached to large depots are now usually washed out with hot water at pressures approximating 100 lb. per sq. in.; this procedure doubtlessly adds to the life of a boiler by reason of the steady temperatures maintained and also ensures much cleaner boilers.

It is customary on modern locomotive boilers to provide a small steam chest, or "turret," on which the various steam connections for the brakes, lubricators, gauges, etc., can be mounted, obviating the necessity for providing a number of independent outlets which are a source of weakness and trouble. A connection by means of an internal pipe is arranged between this turret and the dome, so that it receives supplies of dry steam from about the centre of the boiler.

The pressure gauge generally used is based on the principle of the Bourdon tube, wherein a flattened, curved pipe, when subjected to internal pressure, has a tendency to uncoil or open out, due to the different areas of the surfaces exposed to pressure, inside and outside the curve.

The smoke-box arrangements are preferably simple and few in number, although it is becoming the practice to install some mechanical device for cleaning out the soot, etc., and removing the ashes drawn through from the fire. The blast pipe should be fitted to be as clear of the tubes as possible, and its height regulated to give the best and most effective draft under normal working conditions. As external chimneys have had to be reduced so in height, due to gauge, the common practice now is to provide a very short blast pipe with a "petticoat" to form an extension of the chimney inside the smoke-box. The usual degree of exhaustion in the smoke-box of a locomotive when running is equal to from 2 to 3 in. of water, but at starting it will run up to 6 in., whilst the speed of combustion in the fire-box will mount up to from 50 to 100 lb. of coal per minute—each evaporating 7 to 8 lb. of water per lb.

Nearly all boilers are now provided with a superheater, and the fire-tube type is that generally adopted; it consists of a number of U tubes, extending from a "header" in the smoke-box into large fire-tubes passing through the boiler. The arrangement will be referred to later when dealing with superheater engines.

Seamless steel tubes are used for the superheater elements, the fire-box bends, or ends, being of cast steel, or produced by electric welding, with additional metal introduced. At the smoke-box end, these tubes are attached to a steam collector, which has connections for the steam pipe to the steam chest. The arrangement is such that the steam passing from the regulator to the header must pass through these tubes on its way to the collector steam-pipe and cylinders. As the large fire-tubes in which these elements rest are subject to a flow of heated gases from the fire-box direct to the smoke-box, the steam passing through them becomes highly superheated.

Efforts are now being made to adapt the water-tube system of boiler to locomotives, in expectation that by this means a considerable increase in the speed of steam production will be realized, and this at much higher pressures, without adding to the size and, more important, the weight of the boiler. Examples are in service at working pressures of 350 lb. per sq. in., and upwards; an experimental engine on the Continent uses steam at 850 lb. per sq. in.

MATHEMATICS FOR ENGINEERS

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LESSON XX

SUM AND DIFFERENCE FORMULAE—(*contd.*)

Tan (A + B). In the last lesson we had obtained formulae for *sin* $(A+B)$ and *cos* $(A+B)$, and from them we can obtain the formula for *tan* $(A+B)$, for,

$$\tan(A+B)=\frac{\sin(A+B)}{\cos(A+B)}$$

$$=\frac{\sin A\cos B+\cos A\sin B}{\cos A\cos B-\sin A\sin B}$$

Dividing top and bottom by cos A cos B, we have

$$\frac{\dfrac{\sin A\cos B}{\cos A\cos B}+\dfrac{\cos A\sin B}{\cos A\cos B}}{\dfrac{\cos A\cos B}{\cos A\cos B}-\dfrac{\sin A\sin B}{\cos A\cos B}}=\frac{\tan A+\tan B}{1-\tan A\cdot\tan B}$$

upon cancelling, and remembering that *tan* $=\frac{\sin}{\cos}$. Thus,

$$\tan(A+B)=\frac{\tan A+\tan B}{1-\tan A\tan B}\quad .\quad .\quad (5)$$

and, similarly,

$$\tan(A-B)=\frac{\tan A-\tan B}{1+\tan A\tan B}\quad .\quad (6)$$

If in (5) we put $A=B$, we find

$$\tan 2A=\frac{2\tan A}{1-\tan^2 A}$$

Formulae for Products. Another set of formulae can be found from (1) to (4) of Lesson XIX, which are useful, perhaps not so much at present (except as stepping-stones to those to be given later) as when we deal with Calculus. Adding (1) and (3), we have

(1) + (3)

$$2\sin A\cos B=\sin(A+B)+\sin(A-B)\quad (7)$$

Similarly,

(1) – (3)

$$2\cos A\sin B=\sin(A+B)-\sin(A-B)\quad .\quad (8)$$

(4) + (2)

$$2\cos A\cos B=\cos(A-B)+\cos(A+B)\quad .\quad (9)$$

(4) – (2)

$$2\sin A\sin B=\cos(A-B)-\cos(A+B)\quad .\quad (10)$$

(It was by means of these formulae that the tables were calculated in the first instance.)

Factorizing Formulae. From (7) to (10), another set can be obtained, expressing sums and differences of sines and cosines as products, which are useful in the way that all factorizing formulae are, notably in cancelling fractions, and in obtaining forms to which logarithms can easily be applied. Writing in the above, P for $(A+B)$, and Q for $(A-B)$, when $A=\frac{1}{2}(P+Q)$ and $B=\frac{1}{2}(P-Q)$, these become (on also changing sides)

$$\sin P+\sin Q=2\sin\tfrac{1}{2}(P+Q)\cos\tfrac{1}{2}(P-Q)\quad (11)$$

$$\sin P-\sin Q=2\sin\tfrac{1}{2}(P-Q)\cos\tfrac{1}{2}(P+Q)\quad (12)$$

$$\cos Q+\cos P=2\cos\tfrac{1}{2}(P+Q)\cos\tfrac{1}{2}(P-Q)\quad (13)$$

$$\cos Q-\cos P=2\sin\tfrac{1}{2}(P+Q)\sin\tfrac{1}{2}(P-Q)\quad (14)$$

As an instance when these formulae simplify fractions, we have

$$\frac{\sin P+\sin Q}{\cos P+\cos Q}=\frac{2\sin\frac{1}{2}(P+Q)\cos\frac{1}{2}(P-Q)}{2\cos\frac{1}{2}(P+Q)\cos\frac{1}{2}(P-Q)}$$

$$=\tan\tfrac{1}{2}(P+Q)$$

upon cancelling, and replacing $\frac{\sin}{\cos}$ by *tan*.

Further Triangle Formulae. The formulae developed in the preceding sections enable us to put the formulae of the cosine rule into forms better adapted to logarithmic calculation. First, we will get better formulae for the angles of a triangle in terms of the sides. In this case, we know that

$$\cos A=\frac{b^2+c^2-a^2}{2bc}$$

Now we also have (Lesson XIX)

$$\cos 2A=2\cos^2 A-1$$

or, $\cos A=2\cos^2\tfrac{1}{2}A-1$

so that
$$\cos^2 \tfrac{1}{2}A = \tfrac{1}{2}(1 + \cos A)$$
$$= \tfrac{1}{2}\left(1 + \frac{b^2 + c^2 - a^2}{2bc}\right)$$
$$= \frac{b^2 + 2bc + c^2 - a^2}{4bc}$$

upon using the common denominator, and slightly re-arranging. The first three terms in the above numerator are $(b + c)^2$, so the fraction may be written

$$\frac{(b + c)^2 - a^2}{4bc} = \frac{(b + c + a)\,(b + c - a)}{4bc}$$

on factorizing the difference of two squares. This can be simplified in form by writing s for the semi-perimeter, i.e.

$$a + b + c = 2s$$

when
$$b + c - a = 2s - 2a = 2(s - a),$$
so that

$$\cos^2 \tfrac{1}{2}A = \frac{2s.2(s - a)}{4bc} = \frac{s(s - a)}{bc}$$

Consequently,

$$\cos \tfrac{1}{2}A = \sqrt{\frac{s(s - a)}{bc}}$$

In a similar manner, starting with $\cos 2A = 1 - 2 \sin^2 A$, or $\cos A = 1 - 2 \sin^2 \tfrac{1}{2}A$, we shall obtain

$$\sin \tfrac{1}{2}A = \sqrt{\frac{(s - b)\,(s - c)}{bc}}$$

Now, by division, we find

$$\tan \tfrac{1}{2}A = \sqrt{\frac{(s - b)\,(s - c)}{s(s - a)}}$$

with, of course, similar formulae for $\tan \tfrac{1}{2}B$ and $\tan \tfrac{1}{2}C$. These *tan* formulae are the ones to use in finding the angles of a triangle whose sides are known, when the numbers are too big to do all the arithmetic mentally.

EXAMPLE. Find all the angles of a triangle whose sides are 15·8, 19·6, and 21·4 in. The actual working is as follows—

$$\begin{array}{rcl} a & = & 15{\cdot}8 \\ b & = & 19{\cdot}6 \\ c & = & 21{\cdot}4 \\ \hline & 2) & 56{\cdot}8 \\ s & = & 28{\cdot}4 \end{array} \qquad \begin{array}{rcr} s - a & = & 12{\cdot}6 \\ s - b & = & 8{\cdot}8 \\ s - c & = & 7{\cdot}0 \\ \hline & & 28{\cdot}4 \end{array}$$

$$\tan \tfrac{1}{2}A = \sqrt{\frac{7{\cdot}0 \times 8{\cdot}8}{8{\cdot}4 \times 12{\cdot}6}}$$
$$\tfrac{1}{2}A = 22^\circ\, 32'$$
$$A = 45^\circ\, 4'$$

$$\begin{array}{ll} {\cdot}9445\ (log.\ 8{\cdot}8) & 1{\cdot}4533\ (log.\ 28{\cdot}4) \\ {\cdot}8451\ (log.\ 7{\cdot}0) & 1{\cdot}1004\ (log.\ 12{\cdot}6) \\ \hline 2)\,1{\cdot}7896\ \text{(adding)} & 2{\cdot}5537\ \text{(adding)} \\ \,2{\cdot}5537 & \\ \hline 2)\,\bar{1}\,2359\ \text{(subtracting)} & \\ \,\bar{1}{\cdot}6179\ (5) = log.\ tan\ \tfrac{1}{2}A & \end{array}$$

$$\tan \tfrac{1}{2}B = \sqrt{\frac{7{\cdot}0 \times 12{\cdot}6}{28{\cdot}4 \times 8{\cdot}8}}$$
$$\tfrac{1}{2}B = 30^\circ\, 43'$$
$$B = 61^\circ\, 26'$$

$$\begin{array}{ll} {\cdot}8451\ (log.\ 7\ 0) & 1{\cdot}4533\ (log.\ 28{\cdot}4) \\ 1{\cdot}1004\ (log.\ 12{\cdot}6) & {\cdot}9455\ (log.\ 8.8) \\ \hline 1{\cdot}9455\ \text{(adding)} & 2{\cdot}3978\ \text{(adding)} \\ 2{\cdot}3978 & \\ \hline 2)\,\bar{1}{\cdot}5477\ \text{(subtracting)} & \\ \,\bar{1}{\cdot}7738\ (5) = log.\ tan\ \tfrac{1}{2}B & \end{array}$$

$$\tan \tfrac{1}{2}C = \sqrt{\frac{12{\cdot}6 \times 8{\cdot}8}{28{\cdot}4 \times 7{\cdot}0}}$$
$$\tfrac{1}{2}C = 36^\circ 45'$$
$$C = 73^\circ 30'$$

$$\begin{array}{ll} 1{\cdot}1004\ (log.\ 12{\cdot}6) & 1{\cdot}4533\ (log.\ 28{\cdot}4) \\ {\cdot}9445\ (log.\ 8{\cdot}8) & {\cdot}8451\ (log.\ 7{\cdot}0) \\ \hline 2{\cdot}0449\ \text{(adding)} & 2{\cdot}2984\ \text{(adding)} \\ 2{\cdot}2984 & \\ \hline 2)\,\bar{1}{\cdot}7465\ \text{(subtracting)} & \\ \,\bar{1}{\cdot}8732\ (5) = log.\ tan\ \tfrac{1}{2}C & \end{array}$$

Check—
$$A + B + C = 180^\circ$$

This logarithmic calculation should be quite clear, but there are a few points to notice. First, the choice of the *tan* rather than the *sin* or *cos* formulae is dictated by two reasons; firstly, the tangent can have *any* value, while sines and cosines are always less than 1—and when they are near 1, a small error in their value makes a big error in the angle; secondly, the three formulae for the tangents of the three half-angles include *four* quantities only, as against six and seven for the sine and cosine. Then, in the work, after finding $(s - a)$, $(s - b)$, and $(s - c)$, we added them, and found their sum was equal to s. This *must* always be so, since

$$(s - a) + (s - b) + (s - c) = 3s - (a + b + c)$$
$$= 3s - 2s = s.$$

This fact affords a useful check upon the addition and division. The use of log. tan tables avoids first finding antilogs. and then angles. Also, we have worked out *all three* angles, and then checked them by seeing that they added up to 180°.

Area. The formulae for $\sin \tfrac{1}{2}A$ and $\cos \tfrac{1}{2}A$ above also enable us to get a useful formula for the area of a triangle in terms of the sides. We know already (see Lesson XIX) that

$$\Delta = \tfrac{1}{2}bc \sin A$$

But also (Lesson XIX again)

$$\sin 2A = 2 \sin A \cos A$$

so

$$\begin{aligned} \sin A &= 2 \sin \tfrac{1}{2}A \cos \tfrac{1}{2}A \\ &= 2\sqrt{\frac{(s-b)(s-c)}{bc}}\sqrt{\frac{s(s-a)}{bc}} \\ &= \frac{2}{bc}\sqrt{s(s-a)(s-b)(s-c)} \end{aligned}$$

Using this value of $\sin A$ in the formula for the area just quoted, we find

$$\Delta = \sqrt{s(s-a)(s-b)(s-c)}$$

Example. To find the area of the triangle in the previous worked example.

Using the formula,

$$\Delta = \sqrt{28{\cdot}4 \times 12{\cdot}6 \times 8{\cdot}8 \times 7{\cdot}0}$$

$$= \underline{148{\cdot}5 \text{ sq. in.}}$$

1·4533 (*log.* 28·4)
1·1004 (*log.* 12·6)
·9445 (*log.* 8·8)
·8451 (*log.* 7·0)
2)4·3433 (adding)
2·1716 (5) (dividing = *log.* Δ

Tan ½(B – C). To adapt the other use of the cosine rule to logarithmic calculation, we start from a slightly different standpoint. The case is that in which we are given two sides and the included angle. Knowing one angle, we can immediately obtain the *sum* of the other two: if now we can find their difference, we can find them, and then use the sine rule for the other side.

Starting from the sine rule,

$$\frac{b}{c} = \frac{\sin B}{\sin C}$$

whence

$$\begin{aligned} \frac{b-c}{b+c} &= \frac{\sin B - \sin C}{\sin B + \sin C} \\ &= \frac{2 \sin \frac{1}{2}(B-C) \cos \frac{1}{2}(B+C)}{2 \sin \frac{1}{2}(B+C) \cos \frac{1}{2}(B-C)} \\ &= \frac{\tan \frac{1}{2}(B-C)}{\tan \frac{1}{2}(B+C)} \end{aligned}$$

so that

$$\mathbf{\tan \tfrac{1}{2}(B-C) = \frac{b+c}{b-c} \tan \tfrac{1}{2}(B+C)}$$

with, of course, corresponding formulae involving the other combinations of sides and angles.

Example. As an example of its use, we will solve the triangle in which $a = 28{\cdot}6$, $b = 18{\cdot}5$, $C = 71° 24'$.

Firstly, since $C = 71° 24'$, we must have

$$A + B = 180° - C = 108° 36'$$

so

$$\tfrac{1}{2}(A+B) = 54° 18'$$

Now $\tan \frac{1}{2}(A-B)$

$$\begin{aligned} &= \frac{a-b}{a+b} \tan \tfrac{1}{2}(A+B) \\ &= \frac{10{\cdot}1}{47{\cdot}1} \tan 54° 18' \end{aligned}$$

1·0043 (*log.* 10·1)
·1435 (*log. tan* 54° 18′)
1·1478 (adding)
1·6730 (*log.* 47·1)
$\bar{1}{\cdot}4748$ (subtracting)

whence

$$\tfrac{1}{2}(A-B) = 16° 37'$$

so

$$\begin{aligned} A &= 70° 55' \\ B &= 37° 41' \end{aligned}$$

By the sine rule,

$$\frac{c}{\sin C} = \frac{a}{\sin A}$$

$\bar{1}{\cdot}9767$ (*log. sin* 71° 24′)
1·4564 (*log.* 28·6)
1·4331
$\bar{1}{\cdot}9755$ (*log. sin* 70° 55′)
1·4576 = *log. c*

whence

$$c = \frac{28{\cdot}6 \sin 71° 24'}{\sin 70° 55'}$$

$$= 28{\cdot}69.$$

This result may be checked by using $\dfrac{b}{\sin B}$ instead of $\dfrac{a}{\sin A}$, which gives the same value of c. One point to note in the calculations is that we have not doubled $\frac{1}{2}(B-C)$ to find $(B-C)$, since adding $\frac{1}{2}(B-C)$ and $\frac{1}{2}(B+C)$ gives us B at once, and their difference gives C.

EXERCISE No. 32

1. If $\tan A = \frac{1}{3}$ and $\tan B = \frac{1}{2}$, show that $A + B = 45°$, and find the value of $\tan (B - A)$.
2. Prove that $\sin (n+1)\theta = 2 \sin n\theta \cos \theta - \sin (n-1)\theta$ and that $\cos (n+1)\theta = 2 \cos n\theta \cos \theta - \cos (n-1)\theta$.
3. Remembering that $\cos 60° = \frac{1}{2}$, show that $\sin 70° - \sin 50° = \sin 10°$.
4. Show that $\dfrac{\sin P - \sin Q}{\cos Q - \cos P} = \cot \frac{1}{2}(P + Q)$.
5. Solve the triangles—
   (*a*) $a = 314$, $b = 256$, $c = 188$
   (*b*) $a = 15{\cdot}6$, $b = 19{\cdot}9$, $C = 105°$.

RADIAN MEASURE

Arcs and Sectors. Suppose we want the length of an arc, PQ (see Fig. 88) of a circle of radius r, or the area of the sector OPQ, the angle at the centre of the circle being $A°$. Then the arc is $\dfrac{A}{360}$ of the circumference, and the area of OPQ is $\dfrac{A}{360}$ of the area of the circle, i.e.

$$\text{arc} = \frac{A}{360} \times 2\pi r = \frac{\pi A}{180} r$$

and

$$\text{sector} = \frac{A}{360} \times \pi r^2 = \frac{\pi A}{180} \times \tfrac{1}{2}r^2$$

It will be seen that both of these calculations will be simplified if we have a table of the values of $\frac{\pi A}{180}$ for various values of A. These numbers $\left(\frac{\pi A}{180}\right)$ are proportional to the angles, and really afford another way of measuring angles. To see this, we will find when $\frac{\pi A}{180} = 1$, that is, find the unit in this new way of measuring angles. From the above, we see that in this case,

$$\text{arc} = 1 \times r,$$

or the unit angle is one for which the *arc is equal to the radius*. This is the same angle for *all* circles, and does not depend upon the choice of any particular number, such as 360 or 90.

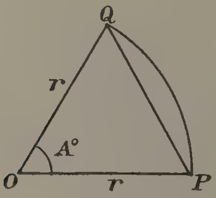

FIG. 88

This angle is called the **radian,** and the measurement of angles in terms of this unit is called the radian measure (or circular measure) of angles. If θ denotes the radian measure of an angle $A°$, we have

$$\theta = \frac{\pi A}{180}$$

$$\text{arc} = r\theta$$

$$\text{sector} = \tfrac{1}{2}r^2\theta$$

and the last two formulae look considerably simpler than those above in terms of A. For converting degrees into radians it is convenient to remember that

$$180° = \pi \text{ radians } \textit{and } 1 \text{ radian} = 57{\cdot}3°$$

as is easily seen from the definition of a radian. Tables of the radian measure of angles are included in most sets of mathematical tables.

In the above, if the radius of the circle is 4 in., and the angle 41°, we find from the tables that 41° = 0·7156 radians, so that

$$\text{arc} = 4 \times 0{\cdot}7156 = \underline{2{\cdot}8624 \text{ in.}}$$

$$\text{sector} = \tfrac{1}{2} \times 16 \times 0{\cdot}7156 = 5{\cdot}7248 \text{ sq. in.}$$

Segment. The area of the segment PQ (Fig. 88) can be found by subtracting the area of the triangle OPQ from that of the sector. Now the area of the triangle OPQ is $\frac{1}{2}r^2 \sin\theta$ (we may use θ instead of A, though this makes no difference to the value of the *sine*). Thus

$$\text{segment} = \tfrac{1}{2}r^2\theta - \tfrac{1}{2}r^2 \sin\theta$$

or,

$$\textbf{segment} = \tfrac{1}{2}\mathbf{r}^2 (\theta - \sin\theta).$$

With the above numerical values,

$$\begin{aligned}\text{segment} &= \tfrac{1}{2} \times 16\,(0{\cdot}7156 - \sin 41°)\\ &= 8\,(0{\cdot}7156 - 0{\cdot}6561)\\ &= 8 \times 0{\cdot}0595 = \underline{0{\cdot}476 \text{ sq. in.}}\end{aligned}$$

EXAMPLE. A chord of a circle of 5 in. radius is at a distance of 2 in. from the centre. Find the area of the smaller segment it cuts off.

In Fig. 89, PQ is the chord, O the centre of the circle, and ON is perpendicular to PQ. We have $ON = 2$ in., and $OP = 5$ in., so that if $\underline{\angle POQ} = \theta$

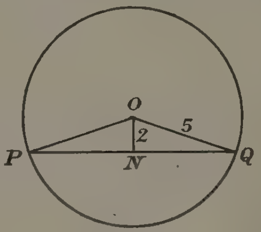

FIG. 89

$$\cos \tfrac{1}{2}\theta = \frac{ON}{OP} = \tfrac{2}{5} = 0{\cdot}4$$

so

$$\begin{aligned}\tfrac{1}{2}\theta &= 66°\ 25' = 1{\cdot}1592 \text{ radians},\\ \theta &= 132°\ 50' = 2{\cdot}3184 \text{ radians}.\\ \sin\theta &= \sin 47°\ 10' = 0{\cdot}7333,\end{aligned}$$

and the area of the segment

$$\begin{aligned}&= \tfrac{1}{2}r^2 (\theta - \sin\theta)\\ &= \tfrac{1}{2} \times 25 \times (2{\cdot}3184 - 0{\cdot}7333)\\ &= \underline{19{\cdot}81 \text{ sq. in.}}\end{aligned}$$

EXERCISE No. 33

1. A fly-wheel is turning at the rate of 110 r.p.m. Express this in radians per second.
2. Find the length of the arc, and the area of a sector of a circle of radius 12 in., the angle at the centre being 75°.
3. Three circles, each of radius 1 in., touch one another. Find the area of the small curvilinear triangle included between them.
4. Find the area common to two circles of radii 4 in., with their centres 6 in. apart.

(*Continued on page* 983)

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XIX

Technical Features in Design. We have, in previous lessons, considered methods of determining the form of a machine part so that the detail may serve its purpose with reliability and efficiency. In order to obtain the above we have given attention to both the strength of the parts and to previous design. We have also observed that whilst most prospective designs have many predecessors of almost similar form, the variation to be made from the reference drawing requires careful attention if we are to produce an efficient design.

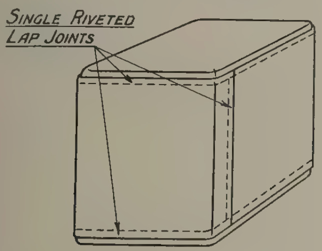

Fig. 1

We may now proceed to consider some technical aspects of design which, being quite independent of the geometry of the form of the part, are not always evident on the drawing.

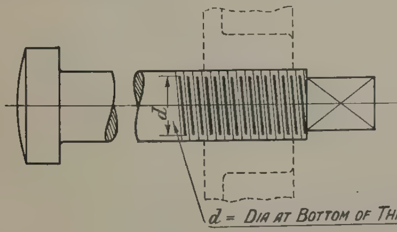

Fig. 3

It sometimes occurs that, as a design develops, certain difficulties of construction are overlooked, and it is a well-known fact that as the draughtsman's attention concentrates upon one point in the drawing the remaining points become hidden from his notice. As a result of this diverting of the critical faculty there is often present, in the making of a drawing, a danger of embodying an impossible feature in the design.

We may proceed to examine a few simple designs possessing various non-technical features.

The Impossible. Fig. 1 shows a steel storage tank made of four plates, two dished-end plates and two side plates, connected by single-riveted lap joints. The flaw in the above design lies in the absence of any provision for the workman leaving the tank after holding up the rivets during the riveting of the last plate. The above omission makes the design, from a constructional point of view, impossible.

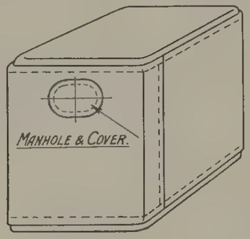

Fig. 2

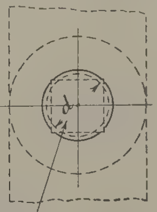

Fig. 4

To make the design practicable we must provide a manhole, as shown in Fig. 2. The above requirement applies, of course, to riveted tanks of all sizes. In the case of tanks of too small a dimension to allow the fitting of either a manhole or a handhole, the areas of the flat plates are sufficiently small to allow welded joints to be substituted for the riveted.

A further example of the impossible in design is shown in Fig. 3, which shows a valve spindle having a square end to take a hand-wheel. The spindle, in the working position, engages in a bridge, and must be screwed into the bridge before being attached to the hand-wheel. The above requirement may, from Fig. 3, appear to be met, but a section of the spindle, taken through the screwed portion, Fig. 4, evinces the fact that the distance across the corners of the square is greater than the diameter at the bottom of the thread. The obvious remedy to make the design practicable is to reduce the square to lie within the circle representing the bottom of the thread in the screwed portion of the bridge.

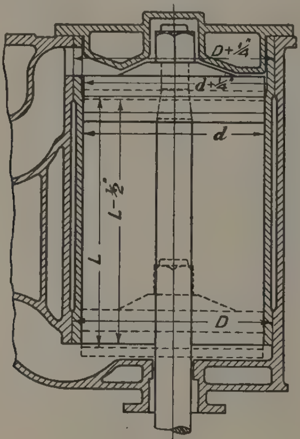

FIG. 5

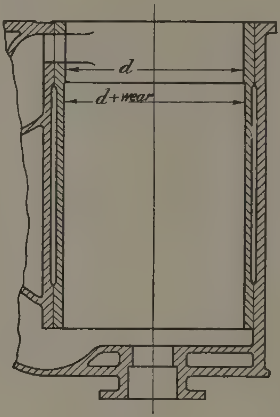

FIG. 6

The Impracticable. Fig. 5 shows a sectional elevation of the high pressure cylinder of an inverted steam reciprocating engine. The piston is shown to full lines at the top end of the stroke, and to dotted lines at the bottom end of the stroke. The length *L* represents the length of surface rubbed by the piston during a complete stroke. Also, the diameter *d* represents the external diameter of the piston rings in the working condition. If, when making the drawing of the cylinder liner we make the internal diameter equal to *d* for the whole length of liner, then after a length of service a certain amount of wear will take place, resulting in an increased diameter within the length *L*. The effect will be to lock the expanding piston rings within the above length *L*. Further, a readjustment of connecting-rod bearing may alter the relative position of the piston at the end of the stroke, with the result that the piston ring would fail to clear the step worn in the liner at the top end. An exaggerated view of the above danger is shown in Fig. 6.

The above objections may be avoided by increasing the internal diameter of the liner by about $\frac{1}{4}$ in. to $d + \frac{1}{4}$ in. from positions just within the rubbing length *L*.

Another technical feature in design is shown (Fig. 5) in the difference in external diameters of the liner. In fitting the liner into the cylinder the lower portions of the liner may pass the upper machined portion of the cylinder body quite freely, whereas if the diameters of upper and lower ends of the liner were similar the

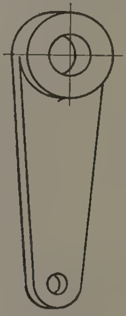

FIG. 7A

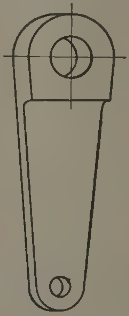

FIG. 7B

liner would have to be pressed into position during the greater part of its length.

The Uneconomical. We have already observed that economy is the leading influence in design. In making a drawing of a new machine part we endeavour to meet the requirements demanded of the part with a minimum of material and labour. Following the above rule we sometimes find it necessary to choose between economy in material and economy of labour. The forged steel lever shown in Fig. 7A has a circular boss at the larger end. This form of boss is more economical of weight for the required strength than the form of boss shown in Fig. 7B, but the latter boss is more easily made and may be considered the more economical design. In practice, the type of boss shown in Fig. 7A would be considered an uneconomical design.

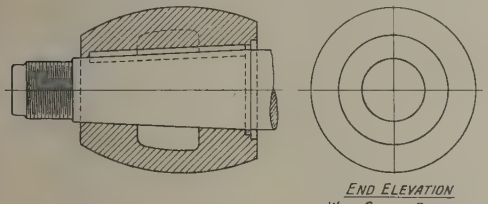

FIG. 8

A further example of the uneconomical in design is given in Fig. 8, which shows the end of a propeller shaft engaged in the propeller boss. It will be seen that the propeller is secured to the shaft by means of a key extending throughout the greater length of the tapered portion of the shaft. If the keyway in the boss is cut to a minimum length, as in Fig. 8, difficulty will be experienced in machining the part for only a portion of the boss length. The alternative method of machining the keyway throughout the whole length of boss, as shown in Fig. 9, is more economical of labour and would be considered a more practical design than Fig. 8.

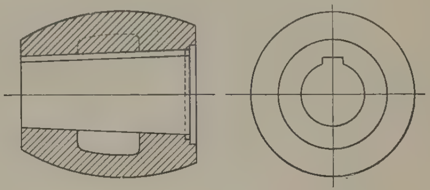

FIG. 9

The above are typical examples of the non-technical features in machine design, and are sufficiently common to suggest the advisability of giving machine drawings a close scrutiny with regard to shop practice before making them working drawings.

MATHEMATICS FOR ENGINEERS

(*Continued from page* 980)

5. An underground passage is 10 ft. wide, and the sides are 7 ft. high. The section of the roof is an arc of a circle of 8 ft. radius. Find the number of cubic yards of earth removed in a length of 20 yd.

ANSWERS TO EXERCISES 30 AND 31

(30)

1. 2·572, 48° 34′, 91° 26′.
2. 6·587, 17° 2′, 22° 58′.
3. (*a*) 7·21 tons, (*b*) 3·576 lb.
4. (*a*) 92° 52′, 38° 37′, 48° 31′,
   (*b*) 104° 29′, 46° 34′, 28° 57′.
5. 53° 19′ and 33° 56′.
6. 47° 10′.
8. 3·857 sq. in., 3·857 sq. in., 14·98 sq. in, 2·905 sq. ft., 9·165 sq. in.

(31)

1. $\frac{4}{5}$, $\frac{12}{13}$, $\frac{56}{65}$, $\frac{33}{65}$, $\frac{16}{65}$, $\frac{63}{65}$.
2. 0·96, − 0·28, 0·352, − 0·936.
4. 0·4472, 0·8944.

APPLIED MECHANICS

By Geo. W. Bird, Wh.Ex., B.Sc., A.M.I.Mech.E., A.M.I.E.E.

LESSON XIX

THE ELEMENTS OF MECHANISM

Moment of Inertia

It is our common experience that no material body has the power of itself to make any change in its own state of rest or motion. For example, the piston of an engine has no power of itself to move along the cylinder, when it does so it is because, either there is applied to it a steam or explosion pressure, or the crankshaft is being turned around by some external force. Similarly, a rotor or fly-wheel, when in motion, is quite unable, of itself, to change its motion by either slowing down or speeding up; such a change can only come about as the result of some external force, such as friction, or a retarding or driving torque. Again, when the rotor or fly-wheel is at rest, it must remain at rest until acted upon by some external force. Now, this *inability* of a body to change its own state of rest or motion we refer to as *inertia*. For a body which is moving or may be made to move in a straight line, this *inertia* may be represented by the *mass* of the body. When, however, we are dealing with a body which may have circular motion, we must take into account the distance of the body from the point about which rotation takes place.

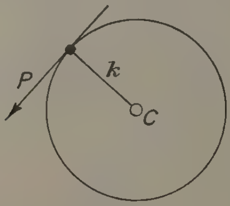

Fig. 102

Let Fig. 102 represent a body, which, under the action of a tangential force P, is rotating about C with an angular acceleration a; the weight of the body being W lb.

Since the angular acceleration is a, the linear acceleration is

$$ak = f \quad . \quad . \quad . \quad . \quad (1)$$

Now, we have already shown that

$$P = \frac{W}{g} \times f \quad . \quad . \quad . \quad (2)$$

$$\text{torque} = P \times k = \frac{W}{g} \times f \times k \quad . \quad . \quad (3)$$

Substituting (1) in (3), we may write

$$\text{torque} = \frac{W}{g} \times ak \times k$$

or

$$\text{torque} = \frac{W}{g} \times k^2 \times a \quad . \quad . \quad (4)$$

Now, assuming that the whole mass of the rotating body be concentrated at its own centre, which is distant k from the centre of rotation, then the quantity $\frac{W}{g} \times k^2$ is termed the *Moment of Inertia* of the body about c, and k is termed the *Radius of Gyration*,

$\therefore$ torque = moment of inertia × angular acceleration,

or

$$\text{torque} = I \times a \quad . \quad . \quad . \quad (5)$$

Centrifugal Force

In Lesson XVIII, Fig. 100, we saw that a body which was moving uniformly in a circle, suffered a change in its velocity, brought about by a change in the direction of its motion. We shall now obtain an expression for the force which acts upon the rotating body to produce this change of velocity.

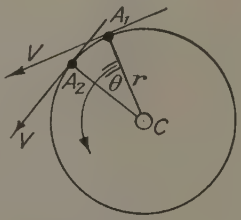

Fig. 103

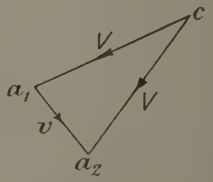

Fig. 104

Let A_1 and A_2 (Fig. 103) be two positions of a body which is moving in a circular path of radius r with a uniform linear velocity V. Let t be the time taken to describe the small arc $A_1 A_2$. To determine the total change in the velocity of the body as it moves from A_1 to A_2, set down ca_1, Fig. 104, to represent the velocity of the body at A_1, and similarly, ca_2 to represent the velocity of the body at A_2. Then a_1a_2 represents the velocity change undergone by the body in moving from A_1 to A_2; that is, a_1a_2 is the velocity which must be added vectorially to the initial velocity at A_1 to give the final velocity at A_2. By taking the two positions A_1 and A_2 nearer and nearer together, the dissimilarity between the sector A_1CA_2 and the triangle a_1ca_2 tends to disappear and, in the limit, when A_1 and A_2 are indefinitely close together, we may write

$$\frac{v}{V} = \frac{A_1 A_2}{r} = \frac{Vt}{r} \qquad (1)$$

$$\therefore \frac{v}{t} = \frac{V^2}{r} \qquad (2)$$

$$\text{But, } \frac{v}{t} = \frac{\text{velocity change}}{\text{time of change}} = \text{acceleration} = f$$

$$\therefore f = \frac{V^2}{r} \qquad (3)$$

To determine the direction of this acceleration f, we note that, as θ is diminished, a_1a_2 which is the velocity change v, becomes more nearly normal to ca_1 or ca_2, and in the limit, when θ is indefinitely small, the direction of a_1a_2 is normal to the direction of motion, that is, the acceleration is radial and directed towards the centre of rotation. Now, to produce this acceleration in a rotating body we require a force which is also directed towards the centre of rotation.

In Lesson XVIII we showed that

$$P = M \times f$$

where P = force acting on a body of mass M

$$\left(M = \frac{W}{g}\right)$$

and f = acceleration of M's motion.
hence, for a body of weight W which is moving with a linear velocity V in a circular path of radius r, the force acting upon the body towards the centre of rotation is given by

$$P = M \times f$$

$$\text{or} \qquad F = \frac{W}{g} \times \frac{V^2}{r} \qquad (4)$$

This force F, is termed the *Centripetal Force*, and since we learned from Newton's third law that *forces occur in pairs*, we must, therefore, recognize the existence of a *Centrifugal Force*, equal in magnitude to the *Centripetal Force*, but acting in the opposite direction, i.e. *radially outwards*.

SIMPLE HARMONIC MOTION

A study of *Simple Harmonic Motion* is of great importance to engineers, since many of the motions in which we are interested are either simple harmonic or close approximations to the simple harmonic type. The piston and plunger of a donkey pump, Fig. 105, move with simple harmonic motion. A weight which is supported by a helical spring, when given a vertical displacement and then allowed to oscillate freely, moves with an approximately simple harmonic motion.

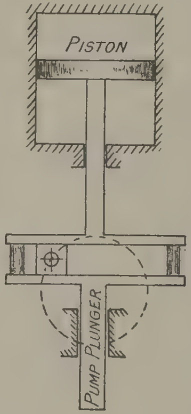

FIG. 105

A bar, fixed to the lower end of a wire which is clamped at its upper end, Fig. 106, moves with, approximately, simple harmonic motion when the wire is set into a state of torsional oscillation. The bob of a long pendulum and several other instances of vibrating bodies may be treated as moving, approximately, with simple harmonic motion. Further, the periodic motion of many parts of machines may be analysed as the sum of a series of simple harmonic motions. From these examples we see the importance, to the engineer, of a study of simple harmonic motion.

Let B, Fig. 107, represent a body which is moving in a circular path of radius a, with a constant angular velocity ω. Then, if C is always the foot of the projector from B upon a diameter, the motion of C will be *Simple Harmonic.* The extreme movement of C from its mid-position is termed the *Amplitude* $= OA$, and the time interval required by C to return to, and to be moving in the same direction through any given point in its path is termed the *Period* $= \frac{2\pi}{\omega}$. Since ω is the angular speed of OB, then $\theta = \omega t$, and

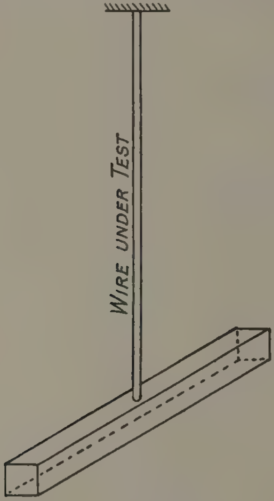

FIG. 106

$$x = a \cos \omega t \qquad . \quad . \quad (1)$$

The linear speed of C is obtained by differentiating (1), and we obtain—

$$v = \frac{dx}{dt} = - \omega a \sin \omega t \qquad . \quad . \quad (2)$$

Differentiating again, we obtain an expression for the linear acceleration of C—

$$f = \frac{dv}{dt} = - \omega^2 a \cos \omega t \qquad . \quad . \quad (3)$$

and, since $x = a \cos \omega t$, (1), we obtain by substitution—

$$f = - \omega^2 x \qquad . \quad . \quad (4)$$

The negative sign in (4) indicates that the acceleration is directed *inwards*, that is, *towards* the mid-position O, whilst x, the displacement, is measured *outwards* from O, and is therefore taken as positive.

When we examine the simple harmonic motion of C a little more closely, we find several points worthy of notice. At each end of its path, C has to come to rest, in order to make the return journey, and so, at the points A and D, the velocity of C is zero. The acceleration f at these points, however, reaches its maximum value, as can be seen from $f = - \omega^2 x$, (4), which has its maximum value when $x = a$.

At the mid-position O, the velocity of C is a maximum, and its acceleration is zero. This can be seen from (2) and (4). When $\theta = \omega t = 90°$, then $v = \omega a \sin \omega t$, (2), becomes $- \omega a \sin 90°$ and its value is $- \omega a$, which is numerically the same as the linear velocity of B. Again, $f = - \omega^2 x$, (4), and when $x = o$ then $f = o$; that is, the acceleration of C is zero when in its mid-position.

TORSIONAL OSCILLATIONS

We can now consider the application of our expressions for simple harmonic motion to the case of a heavy body attached to the lower end of a wire, the upper end of which is fixed whilst the body and wire are given an angular displacement and so set into a state of torsional oscillation.

Since $T = \frac{2\pi}{w}$, where T = time per complete oscillation

$$\therefore \quad \omega = \frac{2\pi}{T}$$

and $$\omega^2 = \frac{4\pi^2}{T^2} \qquad . \quad . \quad . \quad . \quad (5)$$

By substituting (5) in (4), we obtain

$$f = \frac{4\pi^2}{T^2} \times x$$

or $T = 2\pi \sqrt{\frac{x}{f}}$; which may be applied to the case of circular motion and is then written

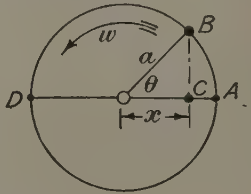

FIG. 107

$$T = 2\pi \sqrt{\frac{\text{angular displacement}}{\text{angular acceleration}}} \qquad . \quad (6)$$

We have previously shown (Lesson VI) that—

$$\frac{f}{r} = \frac{TM}{J} = \frac{N\theta}{l}$$

and we have also shown earlier in this lesson that—

$$TM = I \times \alpha, \quad \text{or} \quad \frac{TM}{I} = \alpha$$

$$\therefore \text{ we may write } \alpha = \frac{N\theta J}{lI} \qquad . \qquad . \qquad (7)$$

By substituting (7) in (6) we obtain

$$T = 2\pi\sqrt{\frac{\theta l I}{N\theta J}}$$

$$T = 2\pi\sqrt{\frac{lI}{NJ}}$$

$$\text{or} \qquad N = \frac{4\pi^2 lI}{T^2 J} \qquad . \qquad . \qquad . \qquad . \qquad (8)$$

Where I = moment of inertia of the heavy body about the centre of oscillation; l = length of wire; N = modulus of rigidity for the material of the wire; J = polar moment of inertia of the cross-section of the wire; T = time of a complete oscillation.

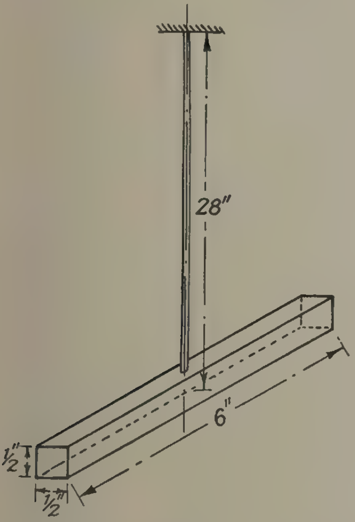

FIG. 108

We have here an expression which is very useful when we wish to determine the *Modulus* of *Rigidity* of any material which can be drawn into a wire of circular cross-section.

EXAMPLE. To determine the value of the modulus of rigidity for a copper-cadmium alloy, a wire of this material was fixed rigidly at its upper end, whilst to its lower end was attached a brass bar 0·5 in. × 0·5 in. × 6 in., Fig. 108. This brass bar was given a small angular displacement and then released, so that it oscillated with a motion that may be taken as *Simple Harmonic*. The weight of the brass bar was 196·7 grms.; the length of the wire under test was 28 in.; the diameter of the wire 0·066 in., and the bar made 109 complete oscillations per minute. Calculate the value of the modulus of rigidity for this copper-cadmium alloy.

We may make use of equation (8) above, and, in order to do this, we must determine the numerical value of the quantities, I, T, and J.

If the student will refer back to Lesson X, he will find that we have there shown that

$$I_{XX} + I_{YY} = I_{ZZ}$$

and we shall now make use of this expression to determine the *moment of inertia* of our brass bar about the longitudinal axis of the wire, that is, about ZZ, Fig. 109.

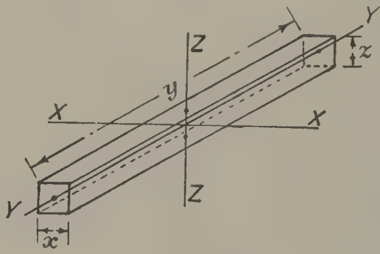

FIG. 109

Let w = the weight per unit volume of the bar, then

$$I_{XX} = \frac{wz}{g} \times \frac{1}{12} \times xy^3$$

Similarly,

$$I_{YY} = \frac{wz}{g} \times \frac{1}{12} \times yx^3$$

$$\therefore I_{ZZ} = \frac{wz}{g} \times \frac{1}{12} xy^3 + \frac{wz}{g} \times \frac{1}{12} yx^3$$

$$I_{ZZ} = \frac{wzxy}{g \times 12}[y^2 + x^2]$$

But $wxyz = W$, the weight of the bar,

$$\therefore I_{ZZ} = \frac{W}{12g}[y^2 + x^2], \text{ and substituting}$$

our values we have

$$I_{zz} = \frac{196{\cdot}7}{12 \times (32{\cdot}2 \times 12) \times 453{\cdot}6} [36 + \tfrac{1}{4}]$$

or $I_{zz} = 0{\cdot}00339$ lb. in. units.

where

W = weight of bar in lb.

g = 32·2 ft. per sec. per sec.

y = 6 in.

x = 0·5 in.

NOTE. In inch units, $g = 32{\cdot}2 \times 12$ in. per sec.^{2}.

The polar moment of inertia of the wire section

$$= J = \frac{\pi}{32} d^4$$

$$= \frac{\pi}{32} \times (0{\cdot}066)^4 = 0{\cdot}000001862 \text{ in.}^4$$

The tine per complete oscillation $= T = \frac{60}{109}$ sec.

The length of wire under test $= l = 28$ in.

On substituting these values in (8), we write

$$N = \frac{4\pi^2 \times 28 \times 0{\cdot}00339 \times 109 \times 109}{\times\ 0{\cdot}000001862 \times 60 \times 60}$$

or $N = 6{,}624{,}000$ lb. per sq. in.

STRUCTURAL STEELWORK

By W. ARNOLD GREEN, M.A., B.Sc., A.M.INST.C.E., M.I.STRUCT.E.

LESSON III

STRUCTURAL STEEL—(*contd.*)

Rivet Spacing. Rivets should not be placed at a closer pitch than three diameters, nor should the centre of a rivet be nearer than one and a half diameters to the edge of the metal through which the hole is made. For a machine-driven rivet, ¼ in. clearance is required between the edge of the rivet head and adjacent projecting metal, such as the leg of an angle or the head of a rivet through it (see Fig. 16).

In order to guard against local buckling in compression members, it is common practice to limit the longitudinal pitch to sixteen times the thickness of the thinnest exposed plate with a transverse pitch double this figure, and to limit the distance of the centre of the rivets to the edge to eight times the thickness of the thinnest plate.

B.E.S.A., No. 153, requires the above longitudinal pitch for tension members and further limits the pitch to 12 in. For compression members, twelve times the thickness instead of sixteen is specified. For angles having two lines of rivets it specifies a pitch along each line of thirty-two thicknesses for tension members, and eighteen thicknesses for compression members, in each case limiting the pitch to 12 in.

Bolts. Where black bolts are used instead of rivets, 20 per cent more bolts than rivets should be employed. This is equivalent to calculating with a shear stress of 5 tons per sq. in. instead of 6. A common allowance for the number of field rivets in a joint is an excess of 15 per cent over shop rivets.

Sometimes turned bolts. which fit tightly in the holes, are used instead of rivets where the positions of the holes makes driving difficult. They have threads of smaller diameter than the bolt which are protected by a sleeve during driving.

Bolts and nuts are usually hexagonal in shape, the thickness of the nut being about the same as the diameter of the bolt and that of the head slightly less. The width across the corners is about twice the diameter of the bolt.

Girder Connections. Just as the sizes of joists have been standardized, it is the general practice for individual firms to standardize connections as much as possible.

Thus, for connecting the end of a joist 20 in. × $6\frac{1}{2}$ in. × 65 lb. per ft. to the web of another girder, a pair of 4 in. × 4 in. angles would be used riveted to the web of the joist by 5 $\frac{7}{8}$ in. rivets with holes in the outstanding legs for 10 $\frac{7}{8}$ in. bolts or rivets, as indicated in Fig. 17. The web of the joist is 0·45 in. thick, so that the bearing value is the determining factor. If 5 tons per sq. in, is the suitable shear stress, the value of each rivet is $10 \times {\cdot}45 \times \frac{7}{8} = 3{\cdot}94$ tons, and the cleat will be good for a reaction of 19·7 tons.

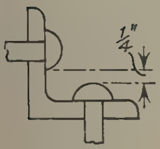

FIG. 16

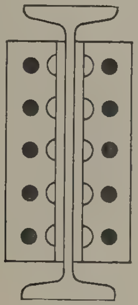

FIG. 17

With a section modulus of $122{\cdot}6 \text{ in.}^3$ and a flange stress of 8 tons per sq. in., the span producing this reaction with a uniformly distributed load is $122{\cdot}6 \times 8 \times 8 \div 39{\cdot}4 \times 12 = 16{\cdot}6$ ft. The same cleats would be used for longer spans with less reactions.

This method of calculation has the merit of simplicity, and is justified by the indeterminable nature of the actual stresses in the rivets.

It is sometimes assumed that the total reaction will act along the face of the web to which the joist is connected, tending to cause the cleats to rotate about the central web rivet.

It is probable, however, that the rivets or bolts connecting the outstanding legs to this web face will provide a certain end fixity for the beam, the deflection in which may tend to cause rotation in the opposite direction.

This uncertainty together with that of the value of the friction grip hardly justifies the complicated calculations sometimes made.

When a joist is carried on a stanchion flange or web, a seating or stool is riveted to the stanchion on which the joist can rest while the connections are completed (see Fig. 18). If the girder cleats are sufficient to carry the load the ultimate structural function of the seating is to increase the stiffness of the connection. An angle cleat riveted to the top flange also serves the same useful prupose.

The stools are often required to transmit all or part of the load from the end of the beam to the column, in which case it is common practice to stiffen them with a tee section or pair of angles as in Fig. 19.

When it is necessary to connect the ends of two joists as, e.g. over the bearings in a continuous length of crane runway, flat plates bolted to each side of the web are used. From analogy with rail connections these are called *fishplates*. Like cleats, they are also standardized.

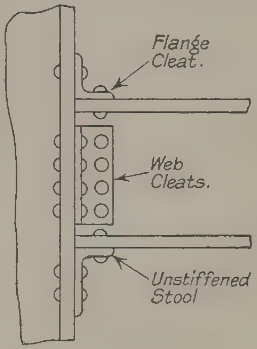

FIG. 18

Though both fishplates and the bolts or rivets through the angle cleats at the end of a girder produce a certain fixity at the support, its value is not considered as relieving the bending moment in the girder at mid span.

Filler Joists in Concrete. If concrete is used as a filling between rolled steel joists for floors, tests have shown that the steel and concrete act together as a composite material with no slipping between the steel and concrete at working loads.

From the fact that the concrete filled up to the level of the top of the joists prevents side flexure of the top flanges, it is usual to permit a higher stress (say, 9 tons per sq. in. instead of 8) on the joists calculated as if acting alone.

If the concrete is 1 in. or 2 in. above the top of the joists, they are sometimes calculated as if stressed when considered as acting alone, to 10 and 11 tons per sq. in. respectively, a rule often justified by the analysis of the section as a composite structure, if the joists are not placed too closely. Breeze concrete should not be employed because of its possible corrosive effect on the steel, and if the higher stresses mentioned above are used the concrete must be good enough to justify the correspondingly high compressive stresses which may occur.

When the depth of the beam carrying the filler joists has to be a minimum, they are carried on a continuous shelf angle riveted to the web of the beam. If the beam depth is unimportant the filler joists can be continuous over the top of the beams.

Compound Girders. If plain steel joists of the allowable depth are not strong enough for a given load they can be reinforced by riveting plates to the flanges.

As the required section modulus is the bending moment divided by the working stress, a diagram of section modulus, such as is shown in Fig. 20, can be constructed. Suppose that the required maximum section modulus is 610 in.^{3}, and it is required to find the width of 3 $\frac{1}{2}$ in. plates riveted to the top and bottom of 2 R.S. joists 20 in. × $6\frac{1}{2}$ in. × 65 lb.

It is easy to show[1] that except for shallow joists the section modulus of two equal plates top and bottom is the area of one plate multiplied by the distance between them.

The total depth of the plated section is 23 in., so that the section modulus contributed by the two joists is × 2 1,226 ÷ $11\frac{1}{2}$ = 213 in. the I of one joist being 1,226.

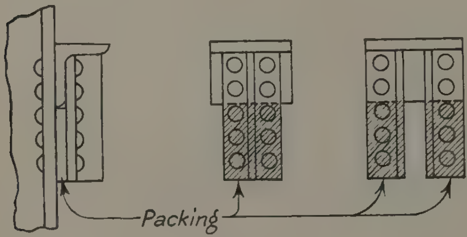

FIG. 19

If ·82 is the thickness of the joist flange at the rivet hole, the area deducted by two $\frac{15}{16}$ in. diameter holes in the flanges and plates is 2 × ($1\frac{1}{2}$ + ·82) × $\frac{15}{16}$ = 4·35 in.^{2} and 20 – (2 × ·82) = 18·36 in. is the distance between them.

The deducted section modulus is thus 4·35 × 81·36 = 80 in.^{3} and the required section modulus of the plates = 610 – 213 + 80 = 477 in.^{3}. The distance between the plates being 20 in., the required area = 477 ÷ 20 = 23·85. The thickness being $1\frac{1}{2}$ in., the width is 23·85 ÷ 1·5 = 15·9. Three 16 in. × $\frac{1}{2}$ in. plates can thus be used.

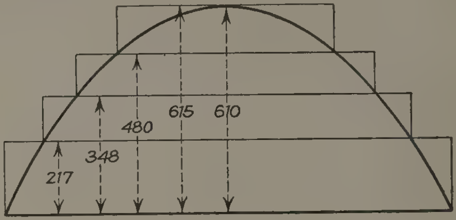

FIG. 20

The section moduli of the two joists with 1, 2, or 3 plates can be found directly by the above method, or by first calculating the moments of inertia as indicated below, slide rule accuracy being well within the rolling margin.

I of 2 20 in. × $6\frac{1}{2}$ in. R.S.J. = 2 × 1,226 2,452

I of 2 rivet holes ·82 × 15/16 = $1\frac{7}{8}$ × (20^3 – $18{\cdot}36^3$) ÷ 12 = 283

Net I of 12 joists 2,169

Effective width of plate 16 in. plate = 16 – $1\frac{7}{8}$ = $14\frac{1}{8}$ in

I of plate 20 in. apart = $14\frac{1}{8}$ (21^3 – 20^3) ÷ 12 = 1,485
Sum = 3,654

,, ,, 21 in. ,, = $14\frac{1}{8}$ (22^3 – 21^3) ÷ 12 = 1,631
Sum = 5,285

,, ,, 22 in. ,, = $14\frac{1}{8}$ (23^3 – 22^3) ÷ 12 = 1,787
Sum = 7,072

Net section modulus with 1 plate = 3,654 ÷ $10\frac{1}{2}$
= 348 in.

,, ,, ,, ,, 2 plates = 5,285 ÷ 11
= 480 in.

,, ,, ,, ,, 3 ,, = 7,072 ÷ $11\frac{1}{2}$
= 615 in.

If horizontal lines at heights corresponding to these values are drawn on the section modulus diagram, the required length and position of the plates are indicated by the intersections. It is usual to carry the plates for about three pitches of rivets beyond the theoretical stopping off points, and to run the bottom plate to the end of the girder.

How the required rivet pitch is calculated will be discussed in the next lesson.

---

[1] See *The Structural Engineer* for Jan. 1923.

OUTLINES OF AUTOMOBILE ENGINEERING

By J. L. Milligan, B.Sc. (Vict.), M.I.A.E.

LESSON IV

ENGINE CONSTRUCTION—(*contd.*)

Cylinder and Pistons

Heat generated by the combustion of the fuel is not all converted into work on the piston, part is lost by radiation to the walls of the combustion chamber and piston crown. This raises the temperature of the parts, and means have to be taken for cooling these to a temperature which will not produce trouble in lubrication or burn pistons or valves. This may be done either by casting ribs on the cylinder and allowing heat to be conveyed to the air, called *direct air cooling*, or by surrounding the cylinder with a water jacket and circulating water through the jacket, the water being subsequently cooled by passing it through a block of piping (the radiator) with a very large surface exposed to the air, which is drawn through it by a fan or driven through by the motion of the car. This is called *water cooling*, although in the case of a car where the water is used again it is really indirect air cooling.

The higher the compression the less the heat lost to the cylinder walls, as more is turned into work. This is not always clearly understood. High compression can cause self-ignition, especially if there is a pointed body or spike of carbon which can become incandescent. Self-ignition produces a harsh metallic sound, called "pinking." There is a well-defined limit of compression for any engine, but two seemingly identical engines may require slightly different values.

A badly designed cylinder may pink at such a low compression value that its compression has to be lowered until it overheats. Overhead valves, owing to the combustion chamber shape, are more free from self-ignition than side valves, and hence can stand higher compression, with less fuel consumption and less heat loss to jackets and pistons.

Aluminium Pistons. Aluminium and its alloys are now almost universal for pistons except in some of the cheapest vehicles. It can, with light weight, convey heat rapidly from the centre of the crown to the part which touches the walls and thence to the jacket, preventing the carbonization of oil on its surface. It has good wearing properties, and its low weight permits of its reciprocation without excessive loads on the crank pin and shaft.

It has a serious drawback: it has a high expansion coefficient. This necessitates leaving the piston small when cold, so that when hot it is a good fit. Rattles when cold are the common lot of aluminium pistons unless special steps are taken to counteract this tendency, by the splitting of the skirt of the piston, fitting of a ring of non-expansible material, or fitting springs inside a loose panel, to name a few of the methods in practical use. Owing to the top of the piston being hotter than the bottom, the top end is made smaller than the bottom.

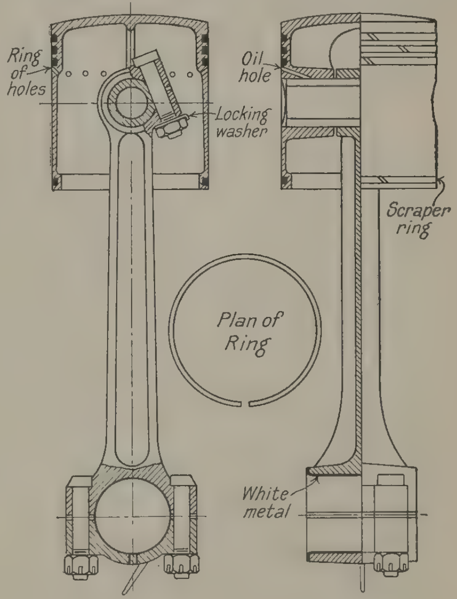

Fig. 18

Piston Rings. In order to keep the piston gas-tight, rings are inserted in grooves in the piston. These are made as Fig. 18, and are not circular when free, but of a shape which when fitted in the cylinder will give approximately equal pressure on the cylinder walls at every point. This is done by different makers in different ways. In use, the gas passes behind the ring and increases the pressure against the wall. There is, of course, a small loss of pressure where the points of the ring come, as they cannot be left touching owing to expansion difficulties, for the piston and its rings heat up more rapidly than the cylinder and cool more slowly. The amount of pressure required to close a ring, called initial pressure, varies with different makers, and can only be settled for a given engine after trial. Excessively high pressures cause wear on cylinder walls. Low pressure rings allow gas to blow down into the crankcase, and may collapse when the pressure in the cylinder is a long way below atmospheric pressure, as during an induction stroke when the throttle valve is closed. This allows air from the crankcase to enter the cylinder and destroy the accuracy of the petrol-air mixture. High pressure rings, by causing friction in the cylinder, require greater force to move them at the stroke ends. This hammering opens out or wears the ring grooves, and causes loss of gas and burning or carbonizing of grooves.

Between these extremes a medium has to be found for ring pressures. Rings for cast-iron pistons are usually made with a width along the piston side of about one-sixteenth of the piston diameter. For aluminium pistons the width is usually half the above.

Specialists in ring manufacture have standardized rings, but when cylinders are rebored and larger pistons fitted, odd sizes have to be made.

The Gudgeon Pin is the hinge on which the connecting rod works. It is fitted in three distinct ways—

1. Fitted non-rotatably in rod, running in the piston without a bush.

2. Fitted non-rotatably in piston, with a bush in the top end of the rod.

3. Fitted rotatably in both rod and piston. Spherical caps are provided on the pin to prevent scoring of the cylinder by end movement.

(*a*) Is the commonest type, and is used both in the cheapest and in the most expensive cars.

(*b*) Is more expensive to produce than (*a*), not so easy to keep quiet and, as the pin is tight in the piston, stresses are introduced in fitting up which are not fair on such a light part.

(*c*) Is a common type, especially where the connecting rod is of duralumin, it avoids stressing the rod head by pinch bolts or keys.

Connecting Rods. Connecting rods are almost all of the same general design (Fig. 18). There are variants, however, rods of circular cross-section drilled out were at one time common, but had a difficulty in the open ends giving insufficient support to the bearing shells.

The big end or crank-pin bearing is lined with an anti-friction lining of white metal. This is sometimes cast directly into the rod, and sometimes into a gun metal shell fitted into the rod. In the former case the whole rod has to be fitted in a special jig and molten white metal poured in, then machined. In the latter case the shells are replaced by new shells already metalled. Remetalling old shells is not an economy, as they will not fit the rod owing to contraction or expansion during the casting process.

On the other hand there is no question that the integral rod end lasts longer than the loose shell type, as heat is more easily conducted along the rod when no joint is interposed. The absence of a shell allows of the valve gear cams being brought nearer to the crank centre.

Small sized rods have two bolts to secure the cap, but four are used on larger sizes. Up to recent years rods were of steel, but duralumin, an aluminium alloy which may be forged and stamped, is in very general use to-day. Stiffness rather than strength is necessary in a connecting rod and weight is undesirable, as many of the stresses in a connecting rod are due to the oscillation and reciprocation of the rod itself.

The influence of rods on "balance" will be later described, but it may be stated that rods should not be less from centre to centre than twice the stroke of the piston.

The big end has to be made with a split bearing to enable it to be put on the crank pin. In a few cases it is not split, and the crankshaft is made up in pieces to enable the rod to be threaded on to the pin. Between the rod and the cap a thin plate, known as a shim, is fitted. When slack develops this may be removed, and one a little thinner substituted or the original filed or ground down.

Connecting rod bolts and nuts must be of the toughest material available, with the threads well cut and a good fit. The breaking of a bolt will do considerable damage, and the restrictions of design do not permit of large bolts being fitted.

NUMBER OF CYLINDERS AND BALANCE

There are two separate requirements in balancing—

(*a*) The power strokes of the various pistons should follow each other at even intervals.

(*b*) The moving parts shall be in dynamic balance.

This can be expressed in a few general principles—

1. The centre of gravity of the whole of the engine parts shall not move. As the crankshaft can be balanced to give a centre of gravity on the centre line and the fixed parts have, of course, a fixed centre of gravity, it follows that the centre of gravity of the group of pistons and connecting rods must not move.

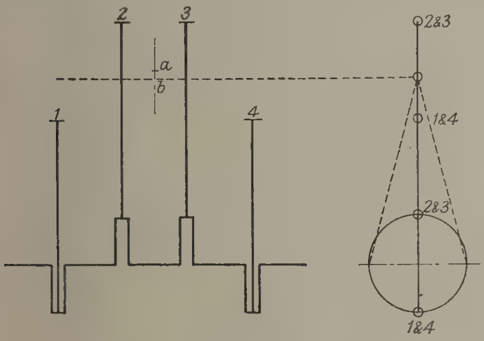

FIG. 19

2. There must be no unbalanced moments. This demands that one-half of the engine must be a mirror image of the other, just as the familiar scheme of half a model ship with a mirror behind gives a ship with one-half the mirror image of the other about a longitudinal plane, the analogy holds in the case of an engine with the mirror transversely to the crank axis at the middle of the engine.

The position and design of the cranks, connecting rods, and pistons should look the same whether a mirror or glass sheet be used. This is probably the simplest way a beginner can understand a subject too often wrapped in abstruse language.

In the case of a four-cylinder engine, we have an arrangement of cranks and rods as Fig. 19. The shaft is balanced and the mirror image theory holds. If *mnop* (Fig. 20) were a mirror, the apparent view of the back half would be the reflection of the front half, which is the same view as seen direct. The two end pistons are up when the two centres are down. The centre of gravity is, therefore, at *a* (Fig. 19) half-way between the centres of gravity of the pistons.

When the cranks are all horizontal the four pistons are all in line, but are then below the point midway between their top and bottom positions owing to the angularity of the connecting rod. This means that the centre of gravity of all the pistons has moved this distance *a*–*b* in a quarter revolution. The next quarter revolution sees the pistons top and bottom again, and the centre of gravity up at its former position.

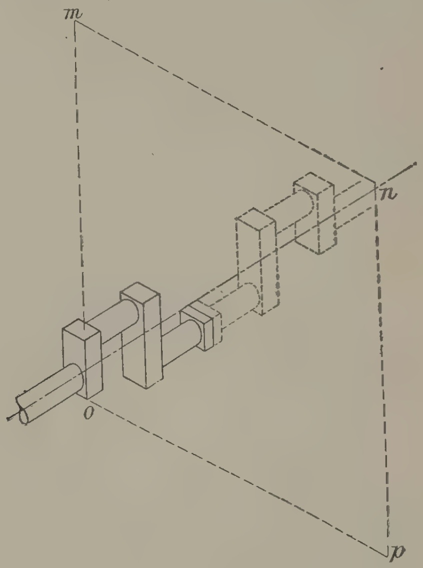

FIG. 20

A weight equal to the combined weight of all the pistons and their gudgeon pins and top end of connecting rods is, therefore, being displaced through a short distance, TWO complete cycles per engine revolution. These are, therefore, called secondary forces.

The distance can be calculated from the known lengths of crank and connecting rod, plotted from drawings, or measured on an engine. When the connecting rod is two strokes in length, the out-of-balance travel is one-sixteenth of the stroke.

As the forces due to reciprocation are proportional to the square of the speed, the four pistons at one-sixteenth of the stroke are equal in their unbalanced secondary force to one of those pistons at the full stroke on primary.

Assuming that the motion of the piston centre of gravity is harmonic (it is nearly so), the usual formula for inertia forces is—

$$\text{Force at each end of motion} = \frac{W}{g} \times 2\pi^2 N^2 S$$

Where W = weight.
g = gravitational acceleration = 32·2 ft./sec./sec.
π = 3·1416.
N = cycles per second.
S = stroke in feet.

Then with a four-cylinder engine whose reciprocating parts, i.e. piston, gudgeon pin, and rod head, weigh 2 lb. each set, 8 lb. in all, with a stroke of 4 in., connecting rod 8 in. centres, at a speed of 3,000 rev. per min., i.e. 6,000 cycles of reciprocation or 100 per second, the secondary stroke is $\frac{4}{16} = \frac{1}{4}$ in. $= \frac{1}{48}$ ft.

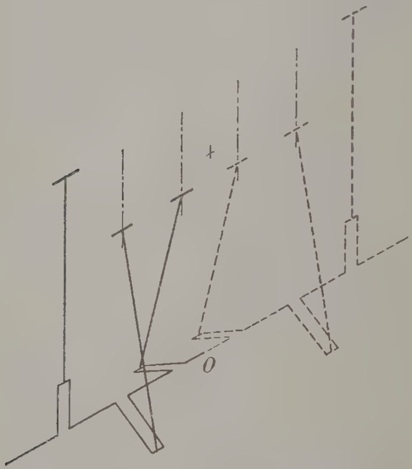

FIG. 21. DIAGRAMMATIC VIEW OF SIX-CYLINDER CRANK

$$\text{Force} = \frac{8 \times 2 \times 9{\cdot}86 \times 100 \times 100}{48 \times 32{\cdot}2} \text{ lb.}$$

approx. = 1,000 lb., which is alternately lifting the engine and pressing it down, twice per revolution.

On a three-cylinder engine, made as shown in full lines (Fig. 21), there is no vertical displacement of the piston centre of gravity, so no unbalanced vertical secondary force, but there is a decided couple, so that if another half engine is added, as shown dotted (Fig. 21), the couple will be balanced by mirror analogy.

There are several ways the cylinders in such an engine can be fired to give even turning, but it is generally arranged that two adjacent cylinders shall not fire in succession.

The elimination, by the use of additional cylinders, of unbalanced forces, has been very marked recently.

The increased length of crankshafts resulting from the increased number of cylinders has given rise to troubles from the shafts twisting slightly and vibrating torsionally whilst rotating, that is, the fly-wheel revolves uniformly, but the front crank moves at varying speeds during its rotation. This produces violent reversals of load on connecting rod bearings, and is known as "thrashing." Stiffness by increasing diameters and web dimensions is the only real preventive, but several cars are fitted with a damper to prevent such vibrations becoming serious.

Crankshafts are arranged in many ways in their bearings. Some have a bearing between each pair of cranks, some have three bearings in four-cylinder engines, some five, some few only two, generally roller or ball in such cases. For six-cylinder engines four and seven are the common types. Some makers fit balance weights to each individual web to relieve the shaft of the bending caused by the centrifugal forces on the crank pins. Some makers have bored out the pins and journals of the crankshafts, but this practice is now almost in disuse, the webs will not stand the removal of the metal and the general adoption of the drilled crankshaft for oil distribution has necessitated the fitting of oil-tight plugs in all the drilled holes, so that the cost is prohibitive and saving in weight very slight.

Crankshafts are usually balanced by revolving at a high speed, and noting the part which is out of balance, putty is added till balance is obtained and metal is removed from the opposite side to the putty, the weight removed being calculated from the weight and position of the putty.

PATTERN-MAKING AND FOUNDRY WORK

By J. McLachlan and C. A. Otto

LESSON X

DIFFERENT TYPES OF PATTERNS—(*contd.*)

We shall now consider a double bend of a different type from that already described. Each flange is parallel with the centre line of the straight pipe. Figs. 38 and 39 explain the method of construction. If such a pipe were only 3 in. or 4 in. diameter, a solid pattern could be more quickly made than a skeleton. Whether the solid or skeleton method is adopted, however, the pattern will be jointed in the same way. In Fig. 40 a vertical section of the mould is shown, and the moulder would make this joint on the line *AB*. The chief difficulty with which he would have to contend would be in getting out the half-flange underneath the bend, and provision would have to be made for this in making the pattern.

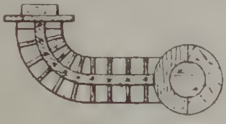

Fig. 38

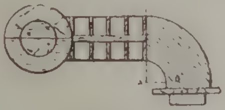

Fig. 39

It is often profitable to make valve-chest bodies by the skeleton method. Large globe valves, for instance, lend themselves to it, if only one or two castings are required. A solid pattern for a large globe valve is very expensive, owing to the shape, which is not very easily made either by the segmental or the lagging method of construction.

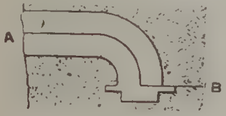

Fig. 40

In making a skeleton pattern for a large globe valve, there is a choice of several methods. The most common is, perhaps, that shown in Fig. 41. Joint-plates or frames are made, and cross-section pieces screwed to them, 2 in. or 3 in. apart. This is a satisfactory pattern up to a point, but a great disadvantage is that there is nothing binding the section pieces on top. This objection is sometimes overcome by fitting strips of timber between the various section pieces. This ensures strength, but it is rather a slow process. There is the further objection to this method that each section piece has to be drawn separately and finished by hand, instead of being drawn from a template and shaped at the sand-papering machine. The lagging principle is the most satisfactory for globe valves. Fig. 42 shows a medium-sized valve made in this way. Joint-plates may or may not be used. If they are used, they form building-plates for the grounds. While this form of construction makes a good pattern, and one which can be very quickly made, it is not suitable for the largest size of valves, because of the width of the longitudinal section pieces

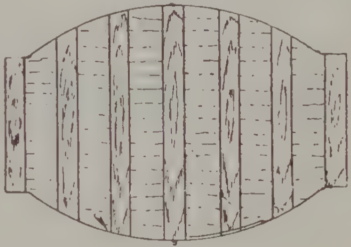

Fig. 41

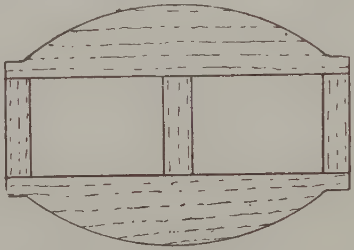

Fig. 42

at the centre ; at the ends the width should be at least an inch.

In Fig. 43 is shown what is undoubtedly the best method of constructing a large globe valve pattern. This is a method which is equally suitable for a solid or skeleton pattern, and it has the great merit of being economical in timber. A stay, as shown, is usually inserted at the joint to bind the grounds together. The grounds, either one thickness or two thicknesses crossed, should be at least 2 in. thick. The section pieces can very conveniently be finished off from a template. It is much better to screw the section pieces to the grounds than to nail them, as is sometimes done.

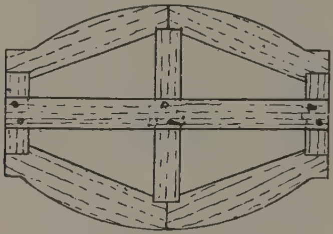

FIG. 43

Pattern Plates. In small work, the use of a pattern plate is essential to cope with the need for rapid production and the reduction of costs. The plate is invaluable when the moulds are prepared by hand, but foundries that make a speciality of repetition work invariably use machines because the rate of production is considerably increased. The special advantage of pattern plates is the cutting out of tool operations that would otherwise be involved in the preparation of each mould. With a proper pattern plate, no tooling whatever should be necessary.

The plates upon which the patterns are fixed may be of wood or metal, or, instead of locating the separate patterns, a cast plate may be prepared with the patterns cast in position. Many foundries use wood plates for small work. They are made to standard dimensions, according to the sizes of the boxes to be used, and have wrought-iron dowel plates set in to suit the box pins. An illustration of a suitable wood plate is shown in Fig. 44. Wood plates, besides being suitable for flat joints, can be cut or have pieces attached to modify the joint to the requirements of the work. Generally, for flat-jointed work, standard cast-iron plates are made and machined all over, to which patterns can be attached. Both these types of plates need to be very carefully lined off, so that the patterns may be set accurately upon them to ensure a correct junction, when the drag and cope are placed together for casting. The patterns to be fixed to these plates may be of wood or metal.

When the work to be done is symmetrical and each box part carries half the mould, or when the whole of the mould can be prepared in the drag necessitating a flat cover, only one side of the plate needs to be used. In the former case, the accurate setting of the half patterns is essential, because both cope and drag impressions are made from the same side, and they must coincide and form the complete mould when brought together. To do this successfully, all plates should have centre lines scribed on them. When the design of the required castings is such that both sides of the plate are necessary, it is generally referred to as a double-sided pattern plate. In this case the pattern sections need to be directly opposite to each other, so that if projected through the plate they would accurately coincide. By drilling holes through the patterns, the pattern sections can be located and fixed on one side, to enable the plate to be drilled, the holes corresponding with those through the patterns. The use of neatly fitting pins will ensure accuracy in locating the corresponding halves of the patterns on the other side of the plate.

Cast Pattern Plates. Many pattern plates are cast along with the patterns as one piece, either in the form of a double-sided or match plate, when each side is used for the cope and drag respectively, or as a reversible plate, when the patterns are carried on one side only, from which both impressions are made. The impressions in this latter form are reversed when assembled for casting. In making this form of plate, a suitable box larger than the outside dimensions of the required plate is procured, and the mould prepared from the patterns in the usual way, greater care, however, being taken to give a very fine surface. Special attention must be given to venting. The runners are

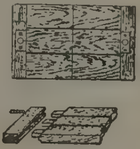

FIG. 44

cut in the manner experience proves to be best for the work in hand. A gate and riser should be prepared for casting the plate. When the pattern moulds are ready, a pattern of the plate is located on the drag and inside the box part. The shape of this plate will depend upon the manner in which subsequent moulds are to be made, whether by hand or machine. For the former, convenient handles are frequently cast on, while a plain rectangular plate is more common for machine use. The thickness of the plate varies, according to its size, from $\frac{1}{4}$ in. to $\frac{1}{2}$ in. When sand has been made up to the thickness of this plate, as in Fig. 45, and runners made, the plate is removed and the mould closed. It must be very carefully clamped before being cast. This type of cast plate may be made of cast iron, brass or aluminium, and pin holes are usually drilled to suit the boxes that are to be used. Many white metal or composition plates are made, however, and, for these, specially prepared frames are used which become part of the plate.

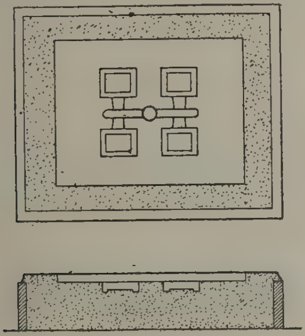
FIG. 45

Reversible Match Plates. The reversible match plate necessitates some form of jig. As the shapes for both drag and cope are required on one side, it is necessary to prepare the moulds from the master patterns in such a way that they can be placed together to form the drag of the plate mould, or partitioned so that each can be cast separately. The former method is suitable for a solid cast plate, and special boxes are necessary. A suitable box is illustrated in Fig. 46, the drag of which is prepared in two equal parts, and flanged at the joint for bolting or clamping together. The joint requires to be machined. It will be noticed that besides the pins and pin holes for registering the box parts as shown, the drag carries separate pin holes for introducing loose pins. With this form of box the drag is, in the first place, separated, and a blank side secured to each part. These then form the two parts for a separate box in which the mould can be made. Room is required about three sides properly to enclose the size of plate with which the patterns are required to be cast. When the mould has been completed, the blank ends are removed and the two parts again secured to form the drag for the plate mould; then the thickness of plate is made up in the same way as in the previous instance. A flat cope is prepared on a ramming board to cover the drag.

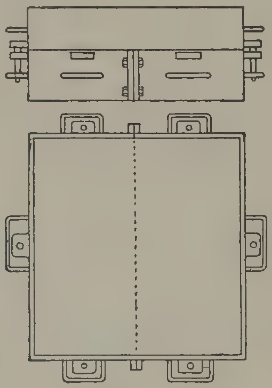
FIG. 46

MACHINE TOOLS

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XVI

BROACHING MACHINES

These are an amplification of the key seating machines that forty years ago invaded the province of the slotting machines. The machines, designed in the first place for cutting key grooves only in bosses with a long serrated tool bar moving horizontally, or vertically, proved so valuable that other sections besides were soon attacked with suitable tools. From these, the broaching machines, more powerful, with much extended utilities and wider applications have been developed. They are of horizontal design, since that is capable of admitting much greater lengths of travel of the broach than the vertical.

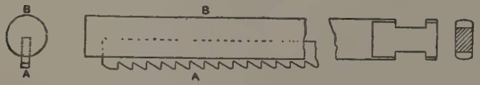

Fig. 108. Serrated Keyway Cutter of Pull Type

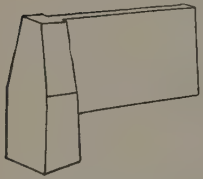

Fig. 109. Slotter Type of Keyway Cutter

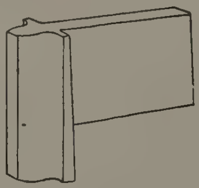

Fig. 110. Rack Tooth Cutter

Hand broaches have always been used, driven with hammer blows to finish holes of square and other sections that have been roughed with drills, chisels, and files. Broaches have the same sections as their holes, and they cut with serrations disposed like the teeth of files, but very coarsely pitched. If a tool of this class is greatly extended in length, and operated with power mechanism the germ of the broaching machine is apparent. But the way of development was prepared slowly by the intermediate forms, the key seaters. Deep key seating on the slotting machines was never satisfactory, either from the aspect of accuracy, or economy. The overhanging tool springs aside laterally, or backwards under heavy cutting, so that the key grooves generally require some correction with the file. For tapered key grooves, the table has to be tilted or wedge packings inserted. Several cuts are necessary to feed into the depth. There is no relief on the upward return stroke, but the tool drags. The diameter of the work that can be taken is limited by the distance between the tool and the framing. Hence, many keyways in large wheels have to be cut with chisel and file. And articles in quantity must be set, each separately on the table by tentative measurements.

The key seating machines are open to none of these objections, and they are made and used extensively in vertical and horizontal designs. The work is supported on a table, or against a plate through which the cutter bar is reciprocated. The space for work is not limited by the interposition of a framing. The cutters used are of two classes—the serrated, Fig. 108, having the teeth *A* gradually increasing in depth in the holder *B*, so that a groove is formed in a single travel of the tool, and the single-edged tool, Fig. 109, clamped in a bar, and then reciprocated, in several successive cuts, feed being given to the work or to the tool. No marking out of the key groove is necessary, as for the slotting machines, but measurements are set by the mechanisms. A great recommendation of the method is, that work is set and centred by the bore, bores of different sizes being accommodated by the fitting of loose bushes to the cutter bar. Other work is done in these machines, cutting rack teeth included, Fig. 110, and work of large dimensions, all leading the way to the more massive broaching machines, with their much extended utilities.

Broaching Mechanism. The work to be tooled abuts against a vertical face plate, directly, or is mounted in a fixture specially designed to hold it. The tool is drawn along through it with a screw travelled with a rotating nut driven through gears from a belt pulley, with quick return; or a motor is used. The length of stroke may range from 21 in. to 8 ft. in the smallest and largest machines, respectively. Cutting speeds may range from

19 to 45 in. per minute, and rates of return from 18 to 50 ft. per minute. There are great differences in the designs of different makers. Hydraulically operated machines are built, but the belt or motor driven geared design is more common, with many variations in the working out of details. A brake may be incorporated to arrest the movement of the screw and prevent it from overrunning when the belt is thrown over between fast and loose pulleys, or an operating clutch pulled. Double machines are built in which the loss of time of the return stroke is minimized by mounting two heads side by side, so that when one is cutting the other is returning.

An Example. Fig. 111 is a photograph of one of the broaching machines by Smith & Coventry, Ltd., of Manchester, with a stroke of 60 in., a cutting speed of 48 in. per minute, and a return speed of 240 in. per minute. The machine is belt driven, with fast and loose pulleys making 650 r.p.m. External views, combined with sections, are given on the plate facing page 1000.

The bed is cast in one piece with a solid bottom to form a trough. Slideways on the inner sides of the bed control the reciprocations of the ram. The operating mechanism is enclosed in the box at the pulley end of the bed. The tube seen extending from the bed protects the end of the screw when in the reverse position, and the attendant from accident. The cutting and reverse strokes of the ram are actuated by large and small gears, the first for cutting, the second for reverse, rotating the phosphor bronze nut that encircles and operates the ram screw. These gears run loosely, but are put into mesh with intermediate gears—compare with the end view to the right—that connect to the first motion shaft with a sliding clutch keyed to the operating nut. The clutch is claw-connected to the large wheel, and cone-connected to the smaller one. To engage these clutches, the wheels are moved along their bearing on the main sleeve, and so sliding together, the movement in one direction, to the right engages the claw clutch for the pulling stroke, and that to the left, the friction cone clutch for the return stroke. Sliding key engagements are thus avoided.

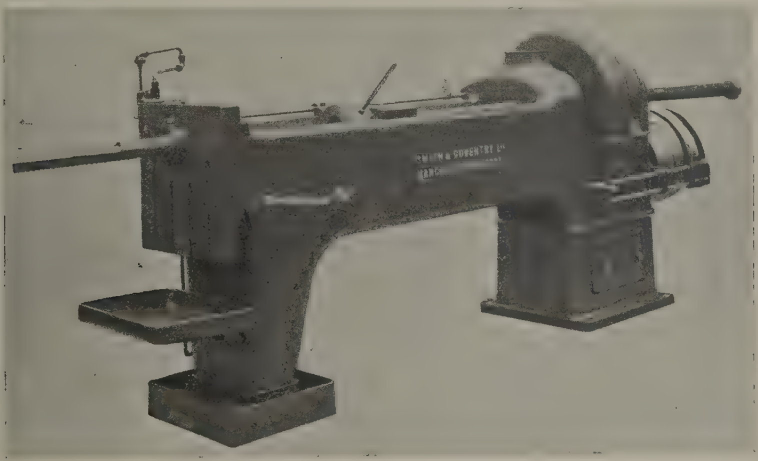

FIG. 111. BROACHING MACHINE

These movements are effected by a lever, seen outside the bed in Fig. 111, towards the left. The connecting mechanism is shown enlarged in Fig. 113. The lever actuates a horizontal link that extends to a lever that moves the sliding clutch. The locking mechanism should be noticed in Fig. 113. The lever is shown in full lines when in the "stop" position where neither clutch is engaged, and is thus held with spring pins thrust against bevelled edges on an extension of the pivoting lever socket. It is similarly held in the "cut" and "return" positions by the action of the spring pins. The stroke is arrested at any portion of its length by means of the adjustable dogs seen on the rod alongside the rear of the machine.

The work is attached to the vertical face of a table at the left-hand end of the machine, seen in Fig. 112, enlarged in Fig. 114. It has milled tee slots for bolts for attaching fixtures, guides, or broach supports. It has a hole large enough to allow the chuck that grips the broaches to be brought nearly flush with the face of the table, so that a minimum portion of the length of the broach is occupied by the shank, and the largest portion left for cutting. The hole is counterbored to receive cast-iron discs with holes to suit the broaches used, or fixtures held with a turned shank. The left-hand end of the broach is gripped in a chuck seen in the upper part of Fig. 113, the head of which slides in the bed. The chuck grips with collets closing in a shallow neck in the shank of the broach, so saving time and avoiding cotter holes. The collets are operated with the vertical lever seen on the sliding head in Fig. 113.

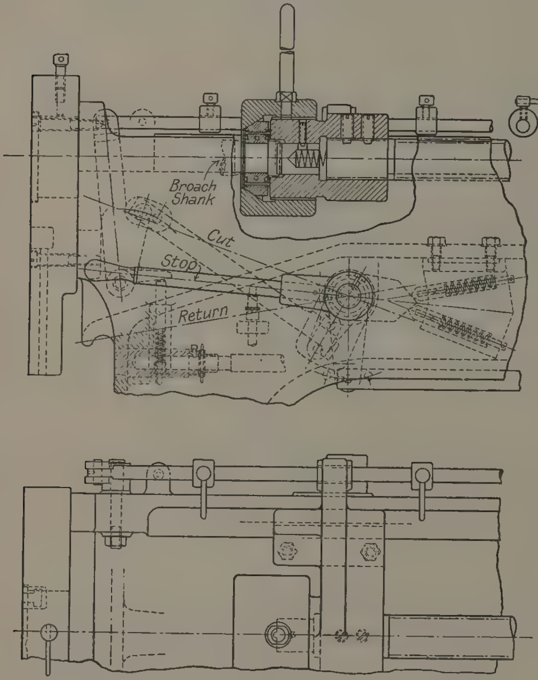

Fig. 113. Arrangement of Cutting Head

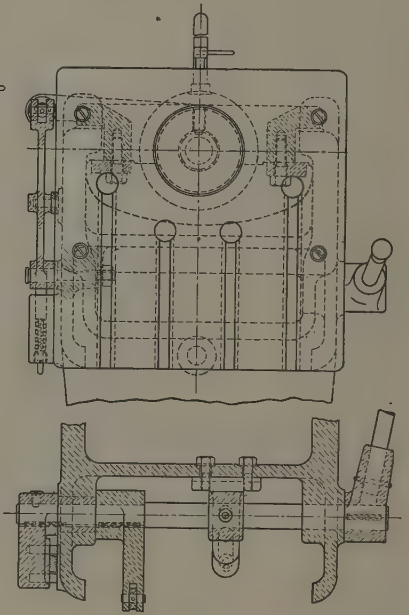

Fig. 114. Details of Head

The pump that supplies cooling liquid to the cutter is seen in Fig. 112, driven with a chain from the left-hand end of the pulley shaft, with provision for adjusting the tension of the belt. A tray beneath the end plate receives the drip, that runs into a tank at the foot of the machine, to be pumped again. The lubricating arrangements are ample. All the running mechanism is contained in an oil bath casing with suitable drips over important lubricating sections. The circulating pump is enclosed in the gearbox. It delivers lubricant through the cutting stroke only. It is of the plunger type, and is operated with a lever, cam bowl, and eccentric cam. The design provides for throwing the pump out of action on the return stroke.

Tools and Operations. The broach is now a high-class tool, made of high speed steel, sometimes of chrome nickel alloy, in lengths frequently of from 3 to 4 ft., and, in exceptional cases, up to 6 ft. long. The rate of feed is varied with materials, and with the pitch of the teeth. A rapid feed with a fine pitch will wear the teeth rapidly, and impart a rough finish to the hole, while too fine a feed will cause rubbing in the hole when the edges become dulled. The

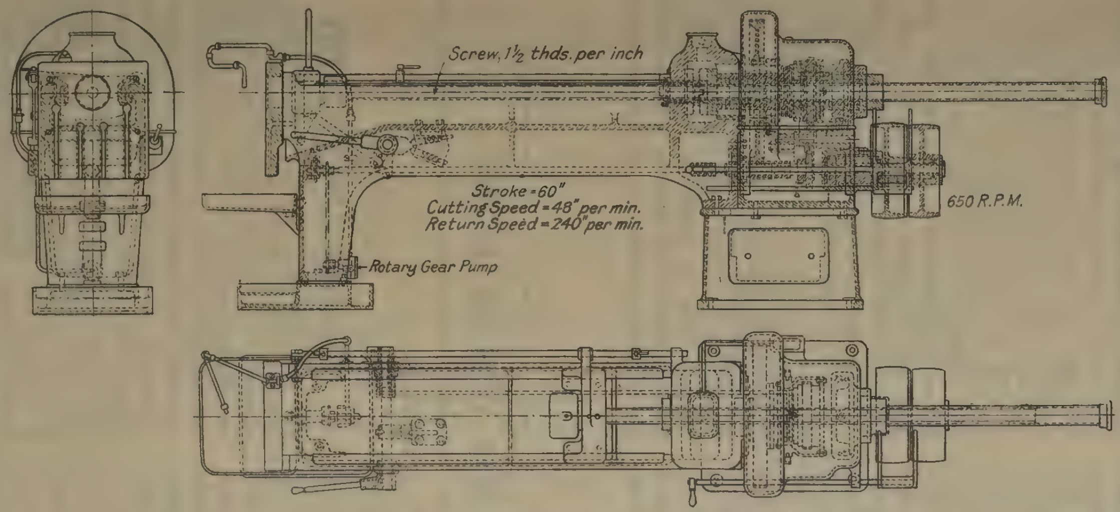

FIG. 112. DETAILS OF SMITH & COVENTRY BROACHING MACHINE

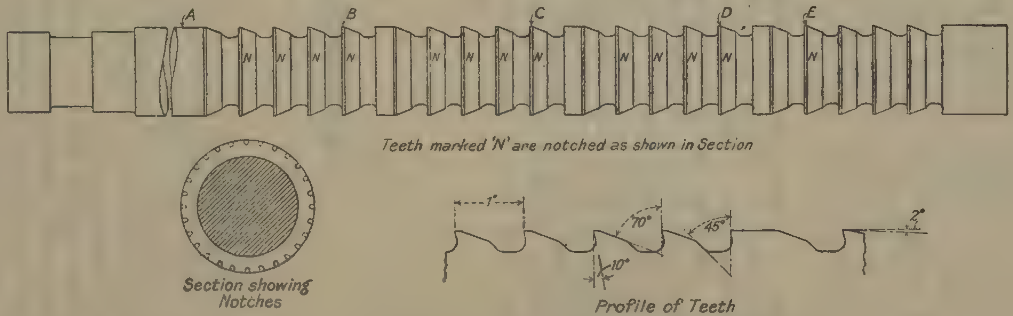

FIG. 115. 2⅝ IN. CIRCULAR BROACH (ARMSTRONG-WHITWORTH)

thickness of chip depends on the feed. An average is 0·001 in., increased to 0·005 in. It follows that much attention is given to the spacing of the teeth, and to their shapes. A coarse pitch is required for hard and tough materials and for long holes, to give ample chip space. This is also favoured by deepening the teeth to prevent pulverizing of the chips. To render the severance of the chips easier, the teeth are sometimes notched, a device which is familiar in some other tools. The sectional forms of the teeth resemble those of saws or milling cutters. An operation may be divided between two, or more tools—roughing and finishing, but these are as commonly embodied in a single broach—the teeth in the front roughing, those in the rear finishing. And these are sometimes detachable to be replaced as they wear, since it is necessary to preserve dimensions within limits.

A remarkable feature of broaching is that it is used extensively for finishing round holes instead of boring them. They work within fine limits and preserve their dimensions. One of these, 40 in. long, is shown by Fig. 115. A section shows the notches in the teeth, which are spaced alternately in adjacent teeth, except in the last three teeth, which, finishing, are left plain. The longitudinal sections of the teeth are shown in the enlarged profile view. The tool roughs and finishes to limits. The work is divided between four sections, separated with plain guide portions, the first three sections roughing, the last one finishing. The guiding portions are ground to 0·0005 in. smaller than the preceding teeth. The diameters at the locations lettered are: At *A*, 2·6105 in., at *B*, 2·615 in., at *C*, 2·6195 in., at *D*, 2·624 in., and at *E*, 2·625 in. Holes are prepared by drilling to within $\frac{1}{32}$ in. or $\frac{1}{64}$ in. of the finished size.

The versatility of the broach is surprising. Spiral grooves are cut with it. Gear teeth on rings, and ratchets are cut. Internal teeth on small rings are cut in one operation, those of larger dimensions are bolted to a revolving index ring. Several may be held in a fixture and cut simultaneously. A great fillip has been given to broaching methods by the employment of splined gears, the grooves in the bosses being broached. Rifled grooves have been produced in this way in guns. Holes in spanners are broached, and in parts of small arms, and other mechanisms.

METALLURGY FOR ENGINEERS

By James Knowles

LESSON VI

THE CRUCIBLE PROCESS

The cementation process, described in the previous lesson, is the oldest method of steel making practised at the present time. Despite the greatest care and skill in the manufacture of blister steel, the selection of the cement bars and subsequent welding operations, however, a certain lack of homogeneity is frequently encountered. It was this lack of uniformity in the quality of the blister steel springs he was compelled to use that induced Benjamin Huntsman, a Sheffield watchmaker, to seek some means of eliminating this source of inconvenience. So successful were his efforts that, in 1740, he invented the crucible process, which is still worked on a considerable scale.

In so far as blister and shear steels, as such, have never been in the fluid state they may be termed steels of plastic origin. The crucible process, on the other hand, consists essentially of mixing and melting dissimilar materials to obtain a desired and homogeneous product. Such reactions as occur during melting are metallurgically unimportant, there being no refining or removal of impurities, except, perhaps, the elimination of involved slag. Hence, for the production of high-class steels for cutting purposes, only the purest materials can be used, for, although poor steel can be made from good ingredients, really good steel cannot be produced from inferior materials.

The best of these are imported from Sweden in the form of white pig and bar iron. Although a good proportion of Swedish Lancashire iron is used, the majority of Sheffield crucible steel manufacturers pin their faith to Walloon iron, notwithstanding its relatively high price. Virtues are claimed for it of which chemical analysis, at least, does not point to its possessing a monopoly, but, so definite are they in this preference that some sound reason for it must exist. This probably lies in the fact that the pig-iron for the Walloon process is made from Dannemora ores which are carefully compounded before charging into the blast furnace and worked on cold blast.

The Melting Hole. A crucible melting shop consists of a series of melting holes ranged down one, two, or three sides of the building, the centre being left clear for casting operations.

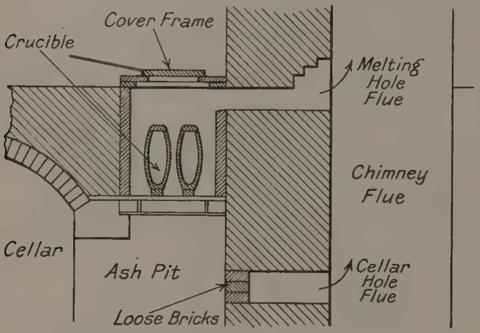

Fig. 6. Diagram Showing the Essential Features of a Coke-fired Crucible Melting Hole in a Sectional Elevation

An individual melting hole is a cubical, firebrick chamber round which gannister is rammed to shape the interior plan to a rough oval with the major axis running towards the chimney stack. The top of the hole is on a level with the shop floor and, during melting, is covered by an iron frame inset with firebrick. Each hole accomodates two crucibles (or "pots") and, being separately connected to the chimney by a flue, is a distinct unit. Thus, any number of holes may be worked and many different "tempers" melted at the same time. Beneath the grate bars of the hole is an ash pit opening into the cellar—also connected to the stack by a flue.

The fuel used is specially selected coke which is broken by hand and carried to the holes in baskets for the fires, which are built up round the pots to lid level. The heat of the furnace is regulated by the simple expedient of placing bricks in the cellar hole flue, or covering it with a sheet of paper. Primitive as these means of draught regulation may appear, it is none the less a fact that perfect temperature control is established by their skilful employment. The individual pots can also be "checked" or "brought on" by slightly raising the cover frame at the back or front as desired.

Crucibles. Pots may be composed of plumbago, graphite, or "white clay," the last named being almost universally used in this country. "White" clay is but a name used for purposes of distinction. No single clay has all the requisite properties and therefore a mixture of several, determined by experience and regarded as a secret, is used along with a proportion of ground coke. This coke, whilst in a fine state of division, is not quite dust, and is an essential ingredient serving a purpose akin to the lattice bars in ferro-concrete. A good mixture for ordinary purposes is composed as under—

Derby clay	48 per cent
Stannington	25 ,,
China	21 ,,
Coke	6 ,,

The crucibles are made by hand on the plant, and it is much regretted that space will not permit of a description of the process of manufacture—a very important adjunct of the practice.

Melting the Steel. The necessary employment of a centring pin in the moulding of the hand-made pots results in a hole in the base which must obviously be filled. The crucibles are accordingly cemented on to stands, when in the furnace, by means of special sand thrown into them to a depth of about 2 in. This, at furnace temperatures frits and sets hard, and thus makes good the base of the pot.

Two clay stands are fixed in each hole and live coals thrown around them. The stands, having been cleaned of cinders, the pots are placed upon them and are carefully fired with coke up to the level of the tops. When a suitable temperature has been attained the sanding is effected, great care being taken that no coke finds its way into a pot meanwhile. A poker test having shown that the bottoms are in good condition, *steeling* is commenced. Cut pieces of Swedish iron and blister bars are first charged, followed by pig-iron and alloys if desired. The steeling is effected by means of an iron funnel, the lower end of which exactly fits the mouth of the crucibles, and the charge is distributed, and its fall broken, by a long poker. The lids are then put on and another fire added. A fire usually lasts about three-quarters of an hour and, for high carbon steels, three are usually sufficient, but the duration of melting also depends on other factors, notably the weight charged and alloys used. Before a fire is applied the remains of its predecessor are worked down with the poker. The third and final charge of coke is known as the *killing fire*, after which the steel is *dead melted*. The crucible is *pulled out* with special tongs, the lid removed and the top of the steel cleared of slag. The teemer, with his tongs, then takes the pot and *teems* its contents into the previously prepared ingot mould.

Killing. An important digression must now be made. It will be remembered that cement bars contain a large volume of occluded gases. If a charge of such material were melted and teemed into an ingot mould these gases would cause the metal to boil over the top of the mould, and any steel remaining therein would be simply riddled with blow holes. The art in melting is to eliminate the gases and so obtain sound ingots. Either of two methods may be employed—

1. Two fires are usually sufficient to get the steel quite fluid, the third or killing fire really achieves a boiling off of the gases in the steel owing to the high temperature, *aided by added manganese*. This, uniting with the oxygen, forms manganous oxide which, reacting with the silica in the pot walls, causes a precipitation of silicon in the boiling metal and, by a complicated chemical reaction, eliminates the hydrogen and nitrogen. The steel is said to be dead melted when the boiling action ceases.

2. The killing action of aluminium is possibly the most remarkable in steel metallurgy. If, to a clear melted steel, a few hundredths per cent of metallic aluminium is added, an ingot can be teemed immediately and will be found absolutely free from blow holes and unsoundness. But the use of aluminium brings another troublesome phenomenon in its wake.

In high carbon and large ingots the centre remains liquid for a considerable time after the sides, aided by the chilling action of the mould, have solidified. The external dimensions of the ingot being thus fixed, internal cavities must obviously be formed by contraction as cooling proceeds. The net result, in the cold ingot, is a funnel shaped cavity at the top, which is known as a "pipe." Precautions taken to reduce this usually entail the use of a preheated fire-clay feeder head, or "dozzle," which, containing about 2 lb. of molten steel, allows a cavity to be made good as soon as it is formed.

Ingot Moulds. Ingot moulds are made of hematite iron in two tongued and grooved portions which, during casting, are clamped together by rings and wedges. Previous to a cast the interior surfaces of the moulds are scoured with hard coke, rubbed over with a rag and "reeked" by placing them, inside down, over the smoky flame of burning creosote. This deposits a coat of soot upon them and obviates any sticking of the ingot to the mould. Crucible steel ingots are usually 3 to 4 inches square, and 56 to 72 lb. in weight; if required larger two teemers, working alternately, maintain a constant stream of molten steel until casting is complete. In this way ingots of over 2 tons in weight are regularly produced.

Gas-fired Furnaces. Modern developments in the crucible process have tended towards the adoption of gas firing. In such plants the melting chambers are built to accommodate about twelve pots each and are heated by producer gas burnt with preheated (regenerated) air. The big advantage of this system is the prevention of sulphur absorption from the coke, but the absolute temperature control over the individual crucibles, obtaining in the old method, cannot be achieved. Hence, only similar tempers can be melted in the same hole.

Methods of Manufacture. These, in conclusion, may be classified as follows—

1. Converted bars selected on carbon contents, melted and killed.
2. Swedish bar iron (which is almost carbonless), plus a calculated amount of charcoal to add the necessary carbon.
3. Swedish bar iron, plus Swedish white pig which gives the desired carbon to the charge.
4. Low carbon acid open hearth, or Bessemer steel scrap, plus Swedish white pig.

Charges are usually calculated so as to include about 15 per cent of crucible ingot scrap of similar carbon content to that of the steel desired.

The last mentioned (4) method is for cheap qualities only. It is by his rigid adherence to the use of pure base materials, plus good crucible steel scrap, that the Sheffield steel maker has built up his reputation and, by sheer purity of product, is enabled, time and time again, "to beat the tariff."

GAS ENGINES AND PRODUCERS

By Arnold Rimmer, B.Eng.

LESSON IX

IGNITION

Various methods of ignition have been applied to gas engines, and may be divided broadly into three classes, viz.—

(*a*) Flame.
(*b*) Hot tube.
(*c*) Electrical.

Flame Ignition

A small portion of gas, ignited from an external source, was carried while still burning, in a valve or slide and brought opposite a port communicating with the cylinder, thus igniting the main charge. This method, fitted on most of the early engines, became obsolete when the slide was found impracticable with the use of higher compression pressures.

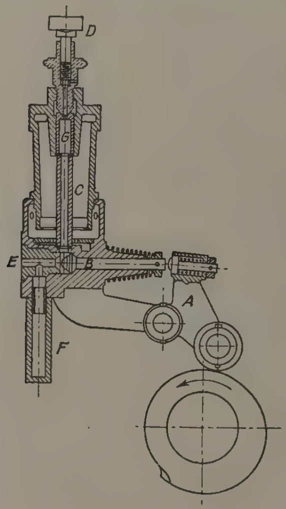

Fig. 46. Hot-tube Ignition Gear (National)

Hot Tube Ignition

A tube of refractory material such as iron, porcelain, or nickel, surrounded by a suitable screen or guard lined with asbestos, is maintained at a bright red heat by a small Bunsen gas flame. The mixture, when compressed, is forced towards the inside of the tube. The instant of ignition is controlled by—

(*a*) Placing the tube at such a distance from the cylinder that the flame just reaches the latter when the piston is at the inner dead centre position.

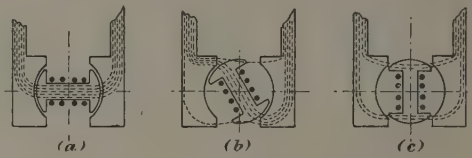

Fig. 47

(*b*) Fitting a mechanically operated "ignition valve" between the tube and the cylinder.

Cleanliness of the connecting passages and of the ignition valve, if fitted, is essential for successful working.

The arrangement as fitted on small National engines is shown in Fig. 46.

At the correct moment the cam allows the lever *A* to move downwards, and the valve *B*, by means of a strong external spring, opens the passage *E* leading to the cylinder. Compressed gases rush into the hot tube *C*, and the resulting ignition is communicated to the cylinder contents. To ensure this action when the connecting passages are somewhat long, an adjustable vent tube *F* and hollow plug *G* are provided, into which the burnt gases left in the passages from the previous explosion may be compressed, while the small relief valve *D* allows the charge to still reach the interior of the tube when the compression is reduced at starting.

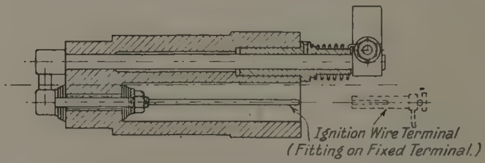

Fig. 48. Low-tension Sparking Plug (Premier)

ELECTRICAL IGNITION

Methods of electrical ignition (which is practically universal on modern engines) may be divided into two main classes, viz.—

1. Low-tension,
2. High-tension.

A low-tension spark is produced when an electrical circuit is broken suddenly, even though the "pressure" on the same is only a few volts. A high-tension spark, however, is one that will actually "jump" across a small gap in an otherwise complete circuit, and this requires a pressure of 2,000 to 5,000 volts, depending on the size of the gap, etc.

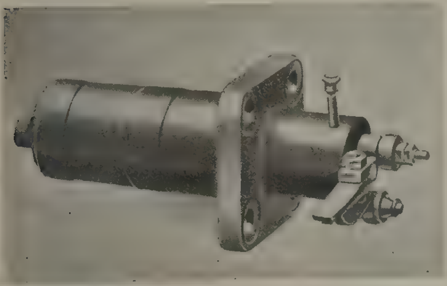

FIG. 49. LOW-TENSION SPARKING PLUG (CROSSLEY)

Low-tension Ignition. The current required for this system is usually derived from a "magneto," which consists of two or more permanent magnets of horseshoe shape, fitted with pole pieces, between which, of course, there exists a "magnetic field." Mounted concentric with these pole pieces is an armature of *H* section, on which a coil of wire is wound parallel to the axis.

When the armature is in the position shown in Fig. 47 (*a*), practically all the magnetic "lines of force" pass through the armature core. In moving through position (*b*) the armature winding cuts across these lines of force and a current is induced in it. This effect will be a maximum if the armature moves to position (*c*). (It will, of course, be understood that the actual magnetic field is here represented by only one or two lines of force.)

The ignition plug consists of a cast-iron body, usually making a ground joint with the breech end of the cylinder. It is held in position by clips or other suitable means. Through it pass two metallic spindles or "electrodes." One of these is fixed and is in electrical connection with the armature winding of the magneto, but is insulated from the body of the plug by mica washers and bushes which also form a gas-tight joint. The other electrode is movable and has a small hammer head (generally of nickel steel) at the inner end which is normally in contact with the fixed electrode. The other end carries a "tappet" lever. In Fig. 48 is shown a section of the "Premier" ignition plug, while that manufactured by Messrs. Crossley, Bros., Ltd., is illustrated in Fig. 49

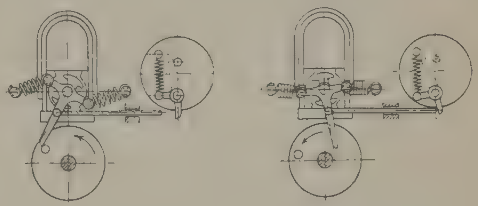

FIG. 50. LOW-TENSION MAGNETO IGNITION

FIG. 51. LOW-TENSION MAGNETO IGNITION (NATIONAL)

A tripping device operated from the camshaft causes the magneto armature to be deflected through an angle of 20-30° from its normal position. It is then released suddenly, and one or more coil springs cause its rapid return. A current is thus generated, which passes to the sparking plug, and at the same instant the tappet lever on the same is operated. This causes the separation of the inner ends of the two electrodes and a spark is drawn out across the gap thus formed. The action is illustrated in diagrammatic form in Fig. 50

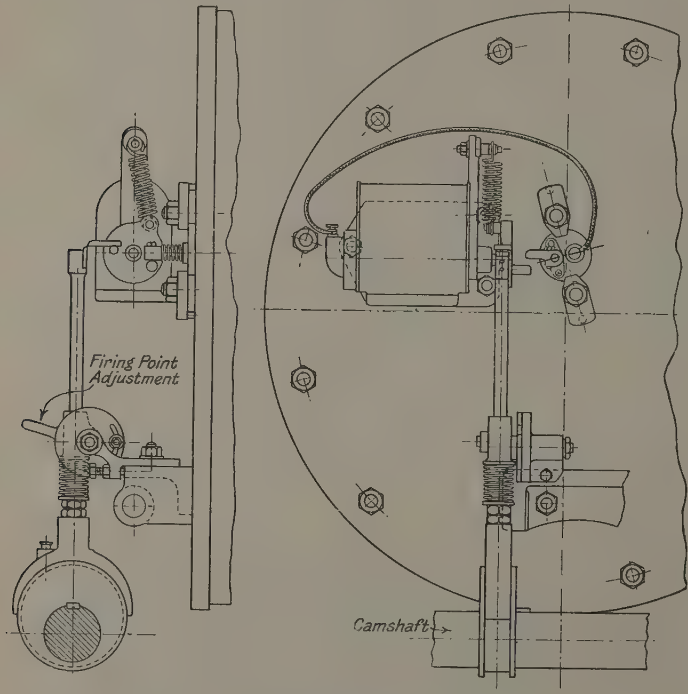

Fig. 52. Low-tension Magneto Ignition (Premier)

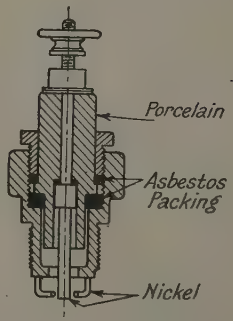

Fig. 53. High-tension Sparking Plug

The low-tension magneto ignition as fitted on horizontal National engines is shown in Fig. 51, and the arrangement on the Premier multi-cylinder engines, in Fig. 52.

High-tension Ignition. This may be effected by—

(*a*) Battery and induction coil,
(*b*) High-tension magneto.

In both cases there are two windings, a primary," consisting of a few turns of comparatively thick wire, and a " secondary," consisting of a large number of turns of much thinner wire. The making and breaking of the primary current at regular intervals causes a current of very high voltage to be induced in the secondary winding, and this passes across the small gap between the " points " of the sparking plug (Fig. 53).

Battery and Coil Ignition. This is illustrated in diagrammatic form in Fig. 54.

A fibre disc *F* is mounted on the engine camshaft, and has let into its edge a brass segment which is in electrical communication with the camshaft. The switch *S* being closed, the primary circuit is completed during the time the segment is passing under the contact piece *P*. This results, of course, in an action similar to the ordinary electric bell, the primary circuit being repeatedly made and broken through the medium of the trembler *T* and contact screw *R*. Thus a series of high-tension currents are induced in the secondary winding, and a shower of sparks passed at the plug, the circuit being completed through the engine frame.

The condenser *C*, fitted across the contact screw connections, strengthens the electrical action of the coil and prevents excessive sparking when the primary circuit is broken. The contact piece *P* is attached to an insulated arm movable about the camshaft, and the point of ignition can be adjusted at will.

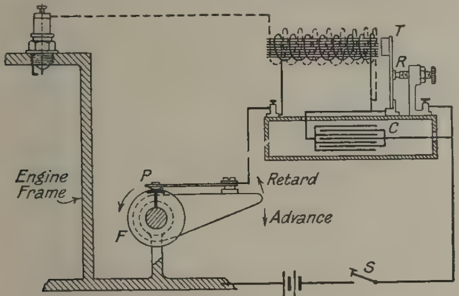

FIG. 54. BATTERY AND COIL IGNITION

On the National vertical tandem engines (described in Lesson VII) the current from low-tension magnetos, which, in this case, have a rotary drive (Fig. 55), is passed through coils to give a high-tension spark.

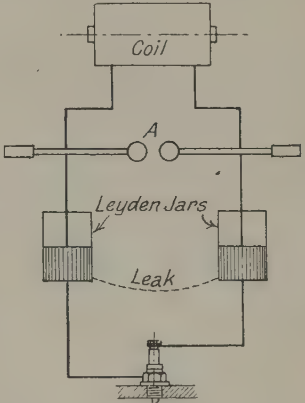

FIG. 56. LODGE IGNITION SYSTEM

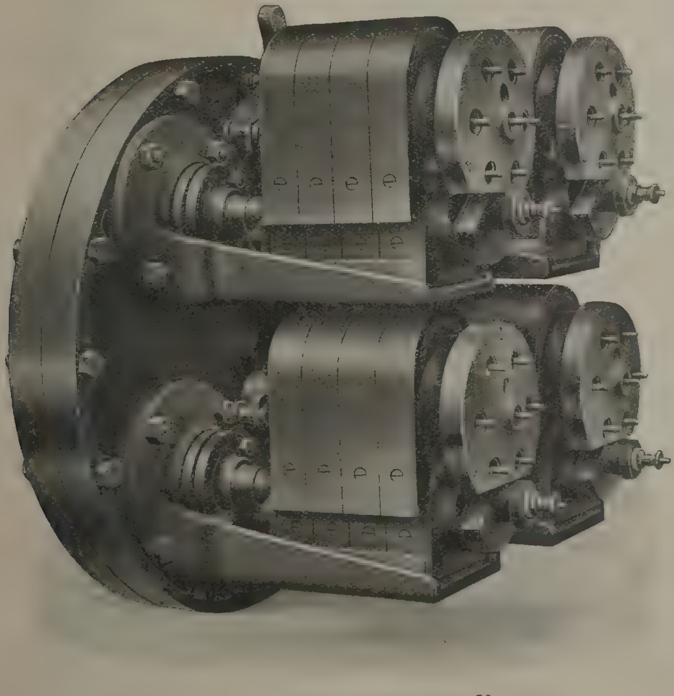

FIG. 55. MAGNETO DRIVE (NATIONAL VERTICAL TANDEM ENGINES)

Lodge Ignition. This is a special form of battery and coil ignition due to Sir Oliver Lodge, F.R.S. It is illustrated in diagrammatic form in Fig. 56.

There is the usual coil and low-tension contact maker on the engine, but the high-tension terminals are connected to the sparking plug through a pair of Leyden jars, the outer coatings of which are short-circuited by a "leak" or imperfect conductor, such as damp wadding in a hermetically sealed glass tube.

A "pioneer" spark gap (kept clean by being under glass) is provided at *A*. Rather than jump this gap, however, the induced high-tension current first flows into the Leyden jars, the leakage path keeping the outer coatings at the same potential. The capacity of the jars being small, however, they almost immediately discharge through the gap *A* and around the circuit of the sparking plug with a "rush of inconceivable rapidity." The leakage path has no time to exert any influence; any dirt, etc., in the path of the spark is flung aside and in something like one millionth of a second, this "B" spark, as it is termed, has passed, though this extremely short time is sufficient for it to ignite the mixture in the cylinder. Owing to "electrical momentum," the B spark is actually of an oscillating nature, and during its discharge the coil takes no further part in the action.

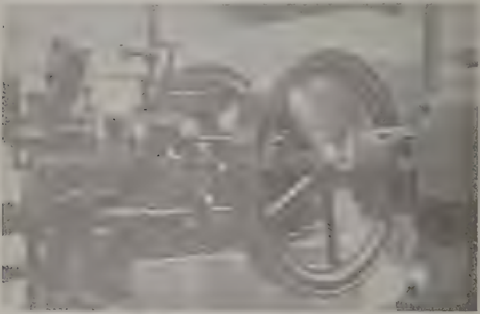

FIG. 57. HIGH-TENSION MAGNETO IGNITION (CROSSLEY)

High-tension Magneto Ignition. The magneto used in this system differs from the low-tension magneto chiefly in the armature, which now has primary and secondary windings. In addition, a "contact breaker" is fitted on one end. By means of the latter the primary current is automatically interrupted at the moment of its maximum intensity, and the induced high-tension current is collected by a carbon brush from a slip ring on the end of the armature and conveyed to the sparking plug. The armature may be either oscillating or rotary, depending chiefly on the speed of the engine. Such a rotary high-tension magneto (as manufactured by Messrs. Hill, Bros., Ltd., Bristol) is shown in Fig. 57.

In this case it is mounted in line with the governor shaft and the connection between the two is through an "impulse starter," which is designed to overcome the difficulty of the weak spark given by the magneto at low speeds. By means of pawls actuated by a rotating cam the armature is held stationary for a brief period. It is then released suddenly and "flicks over" rapidly (under the influence of a spiral spring which has been "wound up" in the interval), thus giving an intense, hot spark. As soon as the engine picks up speed, the pawls are thrown out of action by centrifugal force, and the driving mechanism and armature rotate as a single unit.

In Fig. 58 is shown the arrangement of "Wico" ignition gear as fitted on some of the smaller National engines.

In this magneto the primary current is caused by the collapse of the magnetic lines of force, when the "armature" is pulled away from the magnet cores by the operation of the eccentric fitted on the side shaft. This primary current is interrupted by the opening of the breaker points, and the high-tension secondary current thereby induced passes to the sparking plug.

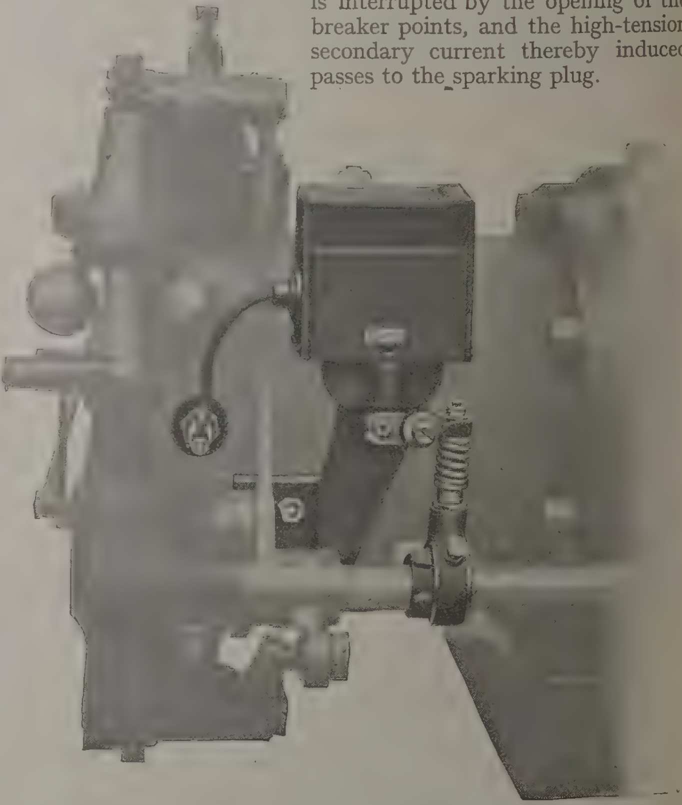

FIG. 58. "WICO" IGNITION GEAR (NATIONAL)

STEAM BOILERS AND THEIR ACCESSORIES

By Percy E. Rycroft, M.B.E., M.I.E.E.

LESSON VII

MECHANICAL STOKERS

Mechanical stoking is a natural development consequent on the great increase in the size of steam-raising plants, the need for securing a maximum economy from the fuel consumed, and the necessity for the prevention of smoke from our factory chimneys.

The utility of hand-fired boilers is obviously limited by these considerations.

If we consider the question of evaporative capacity it will be evident that the evaporation of a boiler depends on the amount of heat which can be liberated beneath it in a given time—in other words, the rate at which fuel can be delivered to the furnace and consumed.

By long experience it was found that the practical limit of " throw " of the average fireman was approximately 7 ft., and, therefore, very few hand-fired furnaces exceeded this distance from front to back. The quantity of fuel which a man can be expected to handle over long periods does not exceed 15 cwt. per hour, so that very definite limits were imposed on the outputs of hand-fired plants by these factors.

With mechanical firing there is no limit to the depth or width of the furnace and no limit to the amount of coal which can be fed into the furnace in a given time, beyond that imposed by the rate of combustion due to draught and temperature considerations.

On the question of economy of fuel consumption, efficient combustion depends on a correct proportion of air being admitted to the furnace at the right time and in the right place.

The even distribution of air through the fuel bed will obviously depend on an even thickness of fire being maintained, and for this to be ensured under hand-firing conditions a high degree of skill is called for from the operator.

This not only involves placing each shovelful of fresh coal in exactly the right position on the fire-grate, but also very careful damper regulation during the process. Furthermore, the admission of a rush of cold air over the fire when the fire door is opened, which at times of heavy load may be as frequently as every five minutes, is in itself a source of inefficiency.

In addition to this, the process of " clinkering " or removal of incombustible material every few hours all mitigates against the possibility of running a hand-fired plant at a maximum and continuous efficiency.

With mechanical stoking, the fire bed can be automatically maintained at an even thickness for any length of time, the fire doors need never be opened, and a perfectly distributed air supply can be constantly and easily secured. Further than this, the process of clinkering is quite automatic and the temperature of the furnace can be maintained at a constant figure over very long and continuous periods.

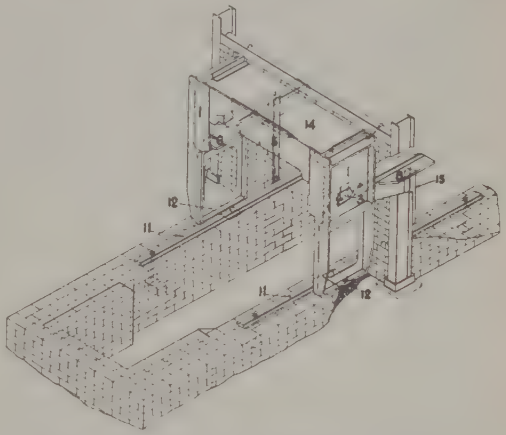

Fig. 41. Brickwork Setting for Chain Grate Stoker

Showing stoker jambs and ashpit

1. Stoker jambs
3. Fire inspection doors
5. Stoker breast plate
6. Stoker arch plate
11. Stoker rails
12. Outside rail plate
14. Stoker lintel
15. Side arch plate support

Dealing finally with the problem of smoke, this is directly caused by the gases given off from the coal under the influence of heat striking against the water-cooled furnace crown or boiler tubes before they have reached a sufficient temperature to properly ignite and combust. This means that very small particles of carbon are carried away in the flue gases and are discharged from the chimney in the form of soot.

With mechanical stoking it is easily possible so to arrange the admission of air, the temperature of the furnace, and the position at which ignition of the "green" coal takes place, as to ensure absolutely smokeless combustion at all times.

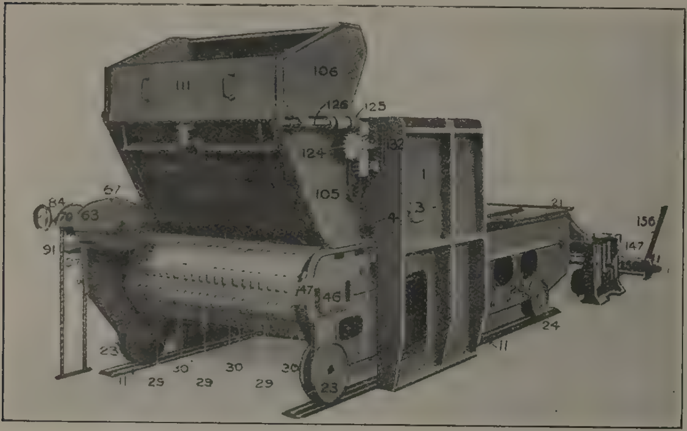

FIG. 42. BABCOCK & WILCOX NATURAL DRAUGHT CHAIN GRATE STOKER

There have been a large variety of devices for mechanical stoking, but the experience of time has now reduced practically all the designs to two main types, viz., "Travelling Grates" and "Retorts."

In the former design a cast-iron endless band, composed of relatively small links or bars, and of the same width as the furnace, is carried on rollers and driving gear at back and front of the furnace.

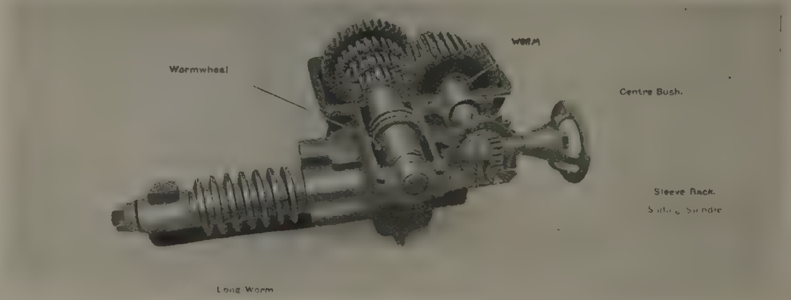

FIG. 43. CHAIN GRATE STOKER GEAR BOX

The fuel drops from a hopper on to the front end of the grate in the form of a ribbon of predetermined thickness, and ignites along its edge immediately it enters the furnace through which it is carried.

By the time the fuel reaches the back end, all combustible matter has been consumed, and only ash or clinker remains. This is automatically deposited over the back end of the grate into an ash dump.

Ignition is obtained and maintained by the provision of a firebrick arch extending for several feet over the front portion of the grate. This becomes incandescent after the furnace has been alight for a short time, and raises the gases given off from the "green" coal to a temperature high enough to ensure smokeless combustion.

In the retort type of stoker the furnace is divided into sections, each retort being a few feet in width and sloping downwards from front to back of the grate.

Coal is introduced at the front end of each section into a channel or retort, and is pressed forward by a plunger or piston, automatically operated by a common crankshaft extending across the full width of the furnace with a connecting rod to each plunger.

Once the grate is entirely covered with coal it will be noted that the fresh fuel is introduced underneath the incandescent fire bed so that the volatiles have to pass through this mass of incandescence—often 2 ft. thick—and are thus sufficiently heated to prevent the formation of smoke.

Air for combustion is introduced into the furnace under pressure through suitable openings in the grate bars.

By a regular movement of these grate bars the clinker is "joggled" to the back, and can be dumped whenever necessary.

In addition to these two main types there are other designs, such as "sprinkler" stokers, which automatically shower the coal into the fire in imitation of hand shovelling; "coking" stokers, by which the fuel is pushed forward from the front by mechanically-operated rams at regular intervals; "screw feed," by which the fuel is fed into the furnace by means of a coarse pitch screw thread running in a trough, and others.

Fig. 44. Locking Handle for Speed Change Gear Handwheel

Some of the latter are still applicable to cylindrical boilers and small plants, but for the bigger installations are now very seldom found in use.

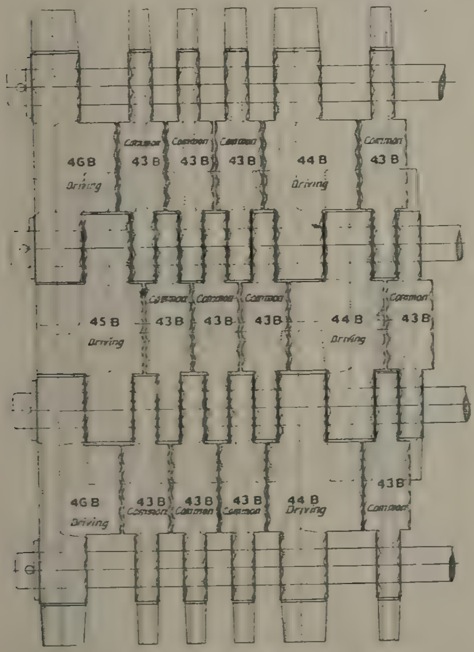

Fig. 46. Stoker Linking Diagram
Showing arrangement of common and driving links

Babcock and Wilcox Chain Grate. The Babcock chain grate is probably the best known example of the travelling grate type and, in its original form, was the forerunner of the numerous modifications which are being manufactured to-day.

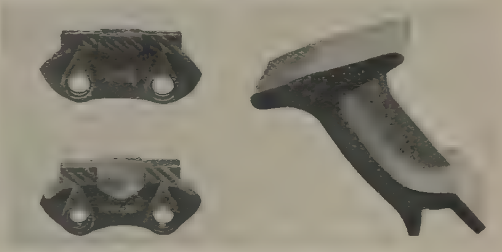

Fig. 45. Links and Ash Plate for Babcock & Wilcox Stoker

In its standard form it consists of a cast-iron framework, at the front end of which is mounted a shaft carrying a number of sprocket wheels more or less equally spaced—about 18 in. apart—over the width of the grate.

Fig. 47. Enclosed Type Chain Grate Stoker
(For natural and balanced draught)

At the back end there is a parallel shaft carrying a drum of similar diameter to that of the sprocket wheels.

Running from end to end of the frame are a series of steel "T" irons, or skids, which support the grate proper and on which the grate links slide.

The grate is composed of cast-iron links of special pattern, and measuring some 9 in. over the ends and a little over an inch at their widest part.

The links have an eye hole at either end, and are interlaced in such a way that steel rods can be passed through them to form a hinged joint between adjacent rows of links, a special feature of the arrangement being that, these joints remain fully closed as the grate passes over the drum and sprockets. The shaft carrying the sprocket wheels is driven by a worm and worm-wheel gearing which is, in turn, operated by an electric motor through change speed or epicyclic gearing arranged to give either four or eight speeds, varying from about 5 to 50 ft. per hour.

Coal is admitted to the grate by means of a hopper extending the full width of the stoker, and fitted with a cut-off shutter.

The thickness of fire is regulated by geared guillotine firebrick lined doors having a range of about 9 in.

At the rear end where the grate passes round the drum it is covered by sectional ash plates, or "dumping bars," which offer sufficient resistance to the advancing fuel to ensure a homogeneous fire bed, but at the same time provide a convenient slope over which clinker is automatically discharged into the ash hopper.

At the front end, to ensure smokeless combustion, a firebrick arch is carried across the full width of the stoker, and extends for varying distances from the front according to the total length of the grate. The sizes of stokers vary from 8 ft. 6 in. to 20 ft. in length, and normally from 3 ft. to 9 ft. in width, although for special conditions they can be built even wider.

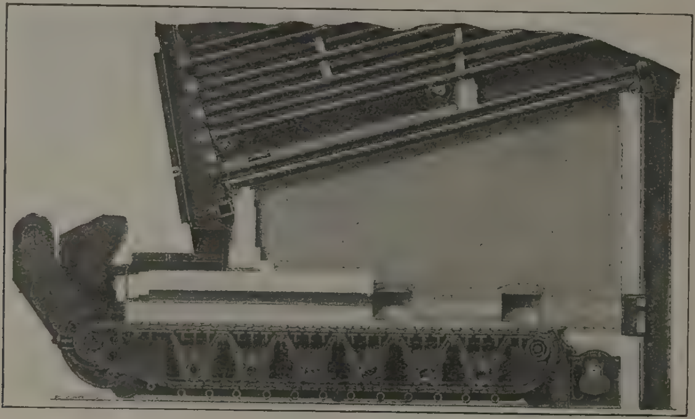

FIG. 48. BABCOCK & WILCOX BALANCED DRAUGHT COMPARTMENT-TYPE STOKER

Under natural or simple induced draught conditions, 25 lb. per sq. ft. of grate area per hour is a normal rate of fuel consumption.

Where high duty is demanded, although the essential features of the grate are maintained, a number of important modifications are introduced.

The space between the upper and lower chains is divided transversely into compartments each about 2 ft. wide. These compartments are hoppered and fitted with regulating shutters to control the admission of air and the release of any fine ash which may fall through the fire bars.

Air is admitted into the compartments under pressure from a common trunk air main, and can be regulated as desired.

By the use of balanced draught with this form of stoker, practically any class of fuel can be burnt and a very high rate of combustion secured.

STEAM CONDENSING PLANT

By John Evans, M.Eng.

LESSON VIII

FEED SYSTEMS AND DEAERATORS

Classification. Feed systems, being the name given to the system through which the condensate passes from leaving the condenser until it arrives at the boilers and economizers, may be divided into two classes—

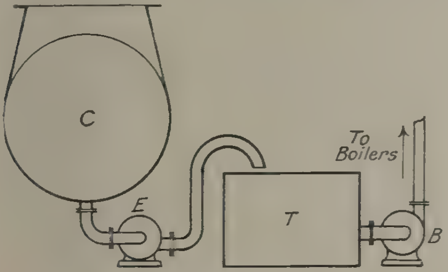

Fig. 38. Open-feed System

(*a*) Open-feed system,
(*b*) Closed-feed system.

The fundamental difference between the two systems is, that in the former the condensate comes into contact with the atmosphere, while in the latter this is prevented.

Open-feed System. Diagrammatic arrangements of both systems are shown in Figs. 38 and 39. The " open " system being simple, is the more common, though not representative of best modern practice.

The condensate, withdrawn from the condenser *C* by the extraction pump *E*, is delivered into an open-feed tank *T*, the boiler feed pump *B* drawing from the tank and delivering to the economizers and boilers.

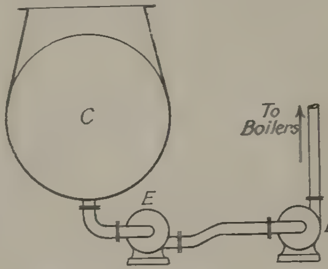

Fig. 39. Closed-feed System

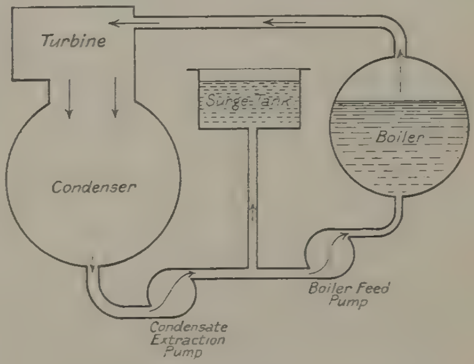

Fig. 40. Simple Closed-feed System with Surge Tank

Closed-feed System. The simplest " closed " system is shown in Fig. 39, where the extraction pump *E* delivers direct into the suction of the feed pump *B*. In practice, however, arrangements have to be made to allow for flexibility of operation between the pumps *E* and *B*, which operate in series. Owing to fluctuations in load, the feed pump *B* may demand more water for the boilers than is delivered by the pump *E*, or it may not require all the water withdrawn from the condenser. To accommodate these fluctuating steam loads, a closed tank is usually installed between the pumps, and a connection made between the tank and an exhaust steam supply at approximately 2–5 lb. per sq. in. gauge. As the pressure in the tank is slightly above that of the atmosphere, ingress of air is prevented. An alternative method of preventing aeration of the boiler feed water is to connect the tank to the condenser by a small bore pipe. With the closed tank there is no air to form an elastic cushion, very rapid condensation of the sealing steam occurs, and, once the tank is filled, it is in effect a large bore pipe. To overcome this difficulty, a *surge tank* is sometimes inserted between the pumps, Fig. 40, in which case there is no circulation of water in the tank except during variations in load, and the water exposed to the atmosphere, will only enter the system during a momentary heavy demand of the boiler feed pump.

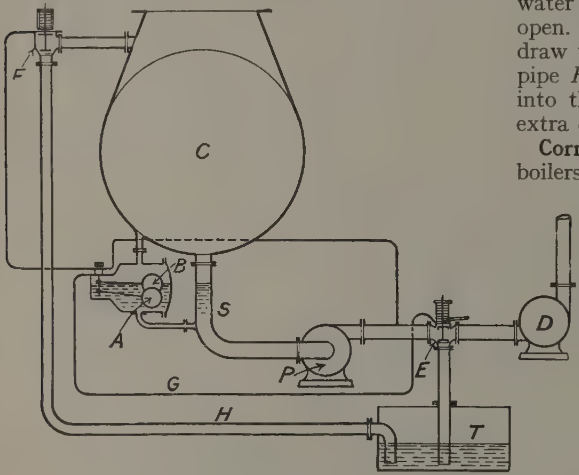

Fig. 41. Weir Closed-feed System for Land Installations

Weir Closed-feed System. The closed feed system adopted by Messrs. G. & J. Weir, Ltd., for land installations, is shown diagrammatically in Fig. 41. Two float-controlled valves E and F are operated by copper floats A and B, housed in a box which is connected both to the condenser C and the suction pipe S of the extraction pump P. The water level in the pipe S is thus the same as that in the float box, and the floats which are set at different levels operate the valves E and F. The top float B controls the valve E which opens as the float B rises, and the bottom float A operates the valve F which opens as the float A falls. When the pump P tends to deliver water in excess of that required by the boiler feed pump D, the water level in S rises, as does the float B which closes its pilot valve. The water pressure in the small pipe G increases, and being connected to the valve E opens it against the action of a spring and allows excess water to pass to the feed tank T. Should the pump P deliver less water than is required by the feed pump D, the float A will fall, open the pilot valve under its control, and, by relieving the water pressure on top of the valve F, allow it to open. The vacuum in the condenser will then draw water from the feed tank T through the pipe H into the condenser, from which it flows into the suction of the pump P to meet the extra demand of the boilers.

Corrosion. How to overcome corrosion in boilers, economizers, and turbines, is one of the greatest problems with which engineers of the present time are confronted, and although they are not agreed on any one of the many theories presented to explain corrosive action, it has been established that the removal of gases in solution in the feed water reduces corrosion troubles by a marked degree. All boiler feed waters contain oxygen, nitrogen, carbon-dioxide and hydrogen in solution, to a greater or less extent, depending on the source of the supply, and the pressure and temperature of the water. The theory of corrosive action which is, perhaps, most generally accepted is that the oxygen, combining with the iron forms iron oxide, which, when reacted on by carbon-dioxide, forms ferrous carbonate. This, on being dissolved in the water, is further reduced by oxygen in solution to iron oxide while the liberated carbon-dioxide, is again free to attack more iron oxide. Thus, the carbon-dioxide remains constant in amount, no matter how small the quantity originally present, and provided an

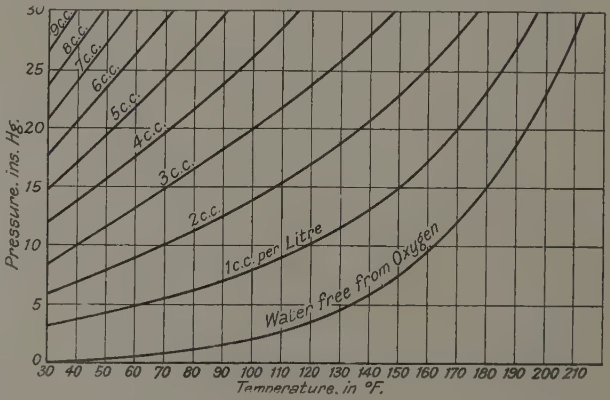

Fig. 42. Solubility of Oxygen in Water

adequate supply of oxygen is maintained the corrosive action is cumulative.

Experiments, and careful study of the reactions, have shown that, in order to prevent corrosion, the oxygen content of the boiler feed water should not exceed 0·1 cub. cm. per litre.

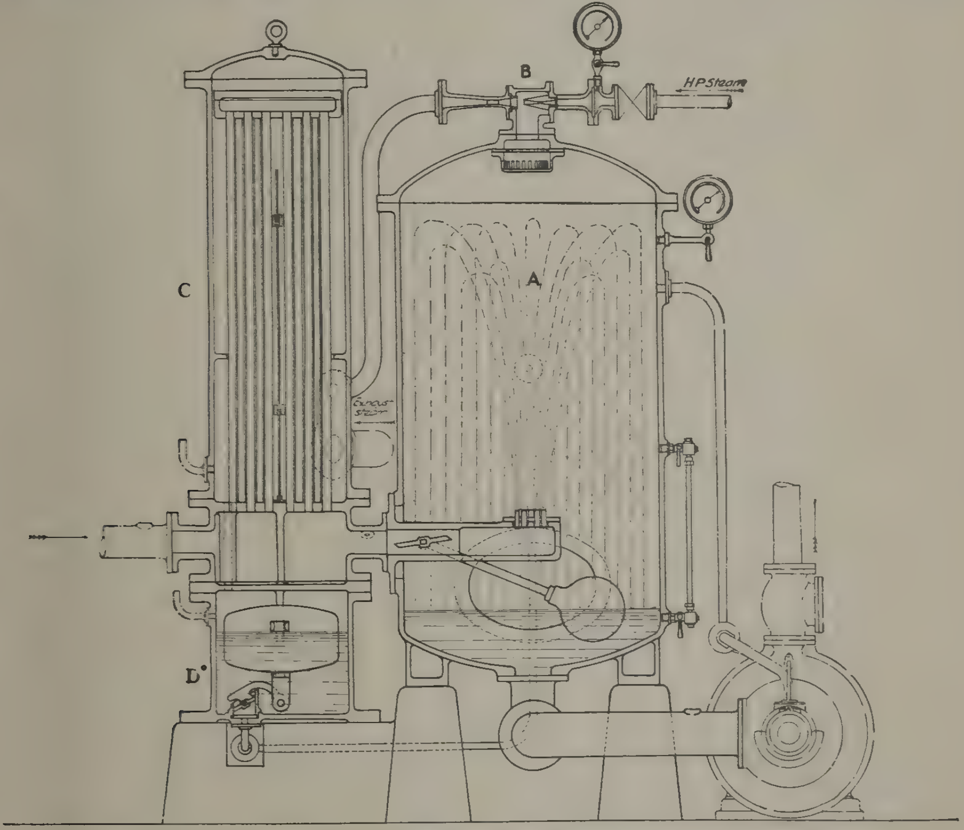

Fig. 43. Arrangement of Hick-Hargreaves Independent Deaerating Plant

An example of the oxygen content of feed water in an open system of a typical power station is as follows. The water discharged from the extraction pump was found to contain 0·9 cub. cm. of oxygen per litre, while half-way through the system it had increased to 3·55, finally leaving the feed pump at 3·75 cub. cm. per litre.

In a modern "closed" system, where the feed water is never in contact with the atmosphere, immunity from corrosion is assured, since the oxygen content seldom exceeds 0·05 cub. cm. per litre, while in the open system corrosion will occur unless the water is "degassed" before entering the feed pump.

Deaerators. The apparatus employed to "degas" or "deaerate" boiler feed water is called a deaerator, and it is becoming increasingly popular, especially as it is easily adapted to "open" systems, and the advantages of the "closed" system achieved at little expense.

The curves shown in Fig. 42 give the equilibrium oxygen content of water at varying temperatures and varying pressures. The obvious method of removing the oxygen and gases is by raising the temperature of the water to the boiling point, but the large amount of heat that would be required at atmospheric pressure would make this a costly operation.

The water is accordingly heated to the boiling point under vacuum, the heat supplied, in addition to being instrumental in removing the gases, increasing the efficiency of the system by feed heating.

The actions of the two chief types of deaerators are as follows—

1. Water is admitted into a chamber in which a vacuum is maintained, and by flowing either through a series of perforated trays or spray nozzles, is broken up into fine "droplets," thus liberating most of the gases in solution, which are removed by an air extractor.

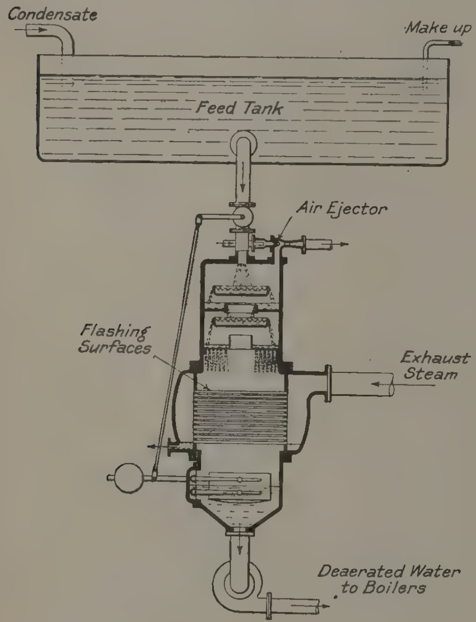

FIG. 44. FEED WATER DEAERATING PLANT

2. Water to be degassed is heated, and injected into a vessel in which the vacuum is greater than that corresponding to the temperature of the water. The sudden reduction in pressure and temperature causes violent ebullition of the water, breaking it up into small particles (flashing), thereby liberating the gases which are removed by an air pump.

An example of the first type is that manufactured by Messrs. Hick, Hargreaves & Co., Ltd., and illustrated by Fig. 43.

The apparatus consists of a deaerating chamber *A*, air ejector *B*, and a surface heater *C*. Water is drawn from the open hotwell through the heater, and thence into the deaerator by virtue of the difference in pressure in the vessel *A* and the atmospheric pressure.

During its passage through the heater the water is raised in temperature to approximately 130° F., by the steam discharged from the air ejector and auxiliary exhaust steam. It is then sprayed in the deaerator by specially designed nozzles which break it up, the liberated air being extracted by the ejector.

The nozzles are arranged so as to spray the water upwards, thus maintaining the water in a finely divided state for the maximum length of time. The deaerated water falls to the bottom of the vessel *A*, and is delivered to the boiler feed pump by an extraction pump. Where the hotwell and deaerator are sufficiently elevated to provide a natural head, the deaerated water will flow direct into the feed pump suction branch. In the base of the deaerator a float valve is fitted, so that in case of a heavy demand for feed water the float will fall and admit more water to the deaerator, the reverse action taking place should the feed water requirements be curtailed.

The deaerator designed by the Contraflo Engineering Co., Fig. 44, is an example of the second type.

Water to be degassed is admitted to the deaerator from the feed tank, a float-controlled valve regulating the supply.

After falling through a series of perforated trays, it falls on the tubes of a surface heater, which are maintained at a temperature higher than that corresponding to the vacuum in the deaerating vessel by exhaust steam. A vacuum corresponding to the temperature of the water leaving is maintained by an ejector. The air and gases in solution in the water are liberated, partly by the "breaking up" process, and finally by violent ebullition on the surface of the heater tubes.

Test for Oxygen Content. Winkler's titration method is usually adopted to ascertain the amount of oxygen present in feed water. Manganous sulphate solution ($MnSO_4$) is first added to the sample under test. By further adding an alkaline potassium iodide solution ($NaHO + KI$) and neutralizing the alkalinity with sulphuric acid (H_2SO_4), the oxygen in the sample is displaced by an equivalent of free iodine. The liberated iodine, dissolved in excess iodide, is then titrated against a standard thiosulphate solution, which enables the amount of oxygen originally present to be determined.

A. Governor.
B. Valve lever fulcrum spindle.
C. Fuel valve.
D. Fuel supply pipe.
E. Camshaft.
F. Arm for moving cams in reversing.
G. Exhaust.
H. Exhaust manifold.
J. Automatic valves for scavenge air supply to cylinder.
K. Lubricating oil main.
L. Cooling-water outlet to cylinder casing.
M. Mirrlees mechanical lubricator.
N. Cooling-water piping to piston.
O. Columns and connecting pipes form reservoir for scavenge air.
P. Lubricating oil overflow to reservoir.
Q. Scavenge air-pumps (double-acting).
R. Air induction piping.
S. Cooling-water inlet to engine.
T. Levers for operating manoeuvring gear.

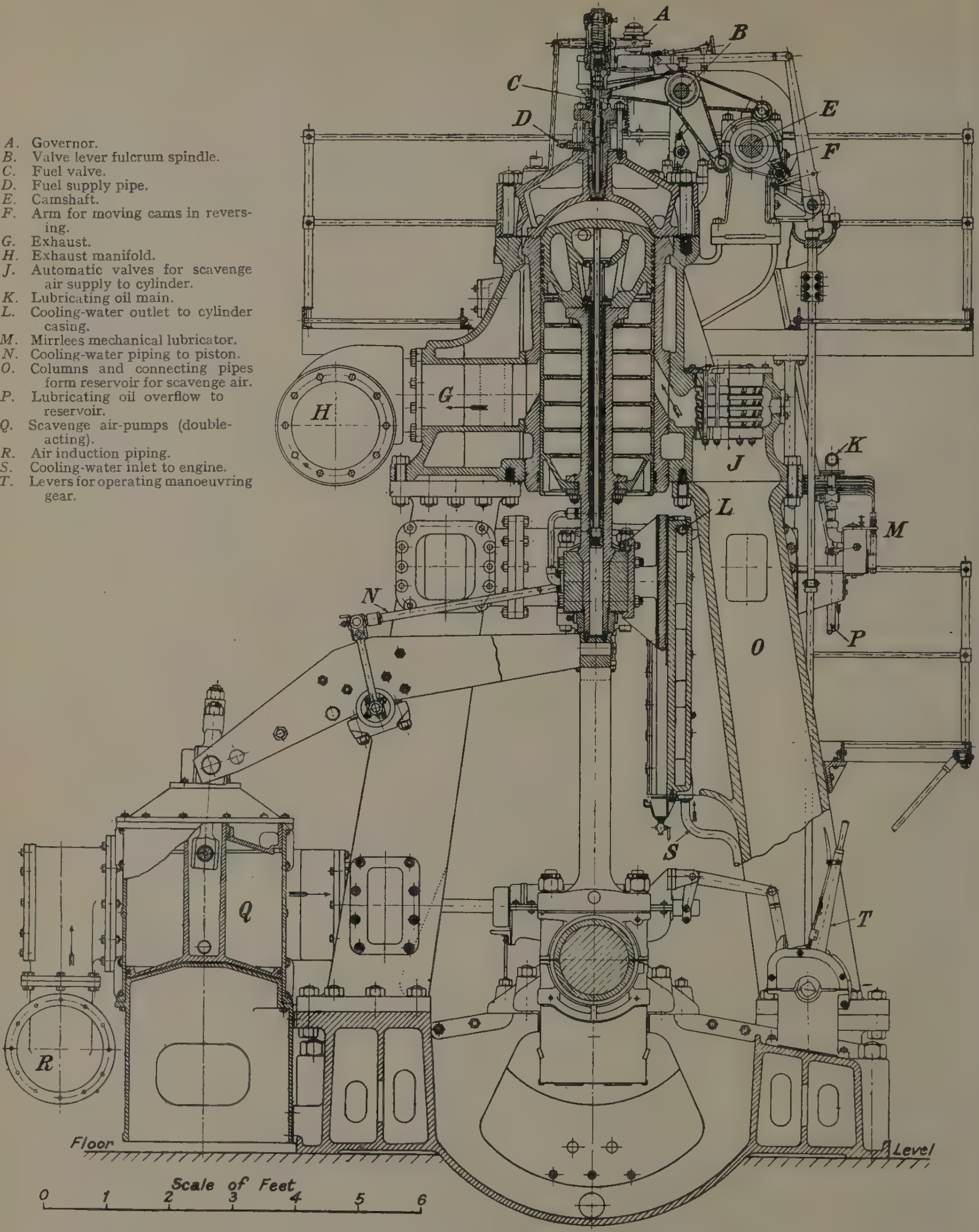

FIG. 35. SECTIONAL ARRANGEMENT OF 1,000 B.H.P. "MIRRLEES-NOBEL" TWO-STROKE CYCLE DIESEL ENGINE

DIESEL ENGINES
CONSTRUCTION, OPERATION, AND MAINTENANCE

By A. Orton, A.M.I.Mech.E.

LESSON IX

MECHANICAL DETAILS OF FOUR-STROKE AIR-INJECTION ENGINE —(*contd.*)

Air Compressors. The blast air is required at a final pressure of about 900 to 1,000 lb. per sq. in., and it is compressed to this degree in two or more stages. In each stage the air becomes heated during compression and it is necessary, therefore, to pass it through a cooler after each stage, so that it may enter the cylinder of the next stage at about its original temperature. By "intercooling" in this way, and by suitably arranging the relative sizes of the cylinders, the maximum temperature attained by the air can be kept approximately the same in all stages. This is important both for the sake of the working parts themselves, and in order to prevent as far as possible, carbonization of the lubricating oil and consequent clogging up of the valves. The valves being invariably of the self-acting or automatic type, it is of the utmost importance to keep them free from any hard or sticky substance. The larger the compressor the less effective becomes the water cooling of the cylinders, and thus it becomes necessary, in order to keep down the final temperature of the air in each stage and to decrease the work of compression, to adopt more than two stages. Few compressors for engines of any considerable size are, therefore, made with less than three stages.

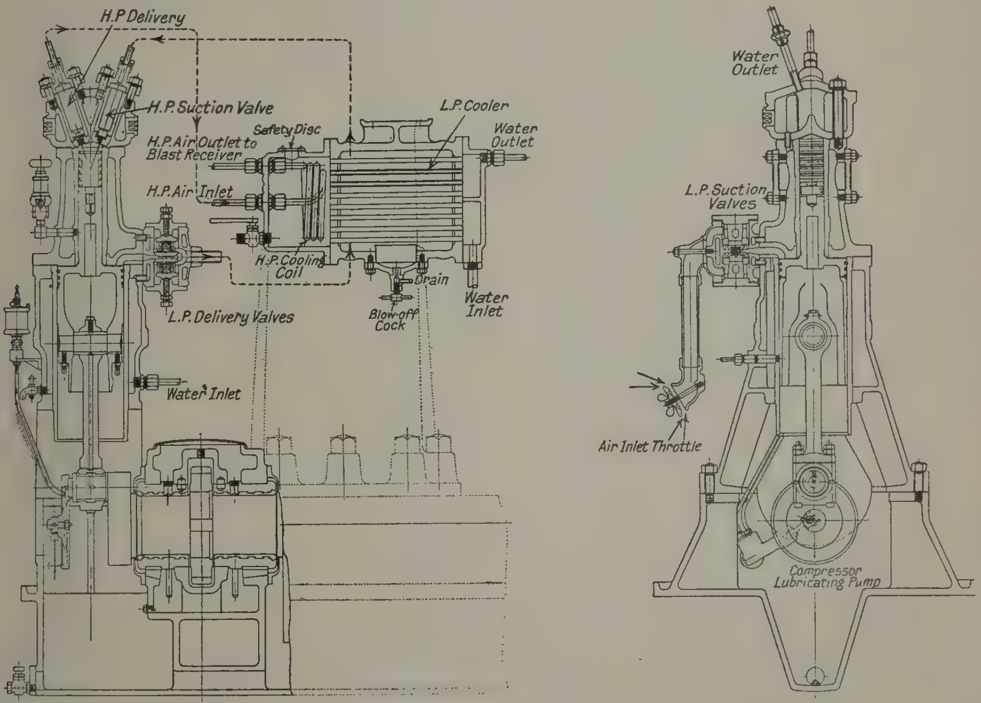

Fig. 32. Two-stage Mirrlees Air Compressor with Coolers

Sectional drawings of a two-stage compressor together with its intercoolers are shown in Fig. 32, the pipe connections being indicated diagrammatically. The figure illustrates one type of Mirrlees compressor, and in both the high and low pressure stages, the suction and delivery valves are plain thin discs of special steel bearings on flat face, each being fitted with a spring to return it to its seat. This type of valve is typical of the present-day practice, though occasionally conical seated valves are still used.

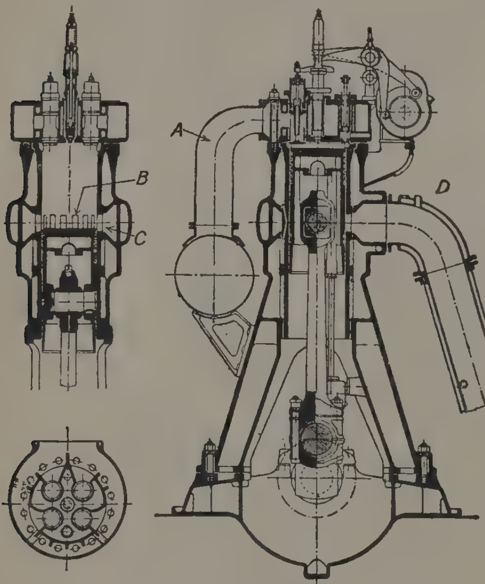

Fig. 33. Normal Two-stroke Valve-scavenging Engine

In this illustration the intercooler is shown as a separate unit from the compressor, but often the two are embodied in one compact construction. Our illustration, however, enables the connections to be rather more clearly followed than would be the case with that type.

Regulation of the air supply is usually effected by a simple throttle valve on the low pressure suction pipe, this being operated either by hand or automatically. A relief valve is always fitted on the delivery system of each stage set to blow off at a suitable pressure somewhat in excess of the maximum normal working pressure of that stage. In the first and intermediate stages this relief valve is fitted on the intercooler, while in the final stages it is usually fitted on the blast air receiver.

Starting and Blast Air Receivers. The starting air receivers, of which there are two, one being a standby, are simple steel vessels, each of such a capacity as to provide sufficient air to start the engines a reasonable number of times without refilling, thus providing against the possibility of the engine failing to start at the first attempt owing to some mishap or incorrect procedure.

Each air receiver is fitted with a steel head containing the control valves, i.e. the main valve admitting air to the starting valves, a small valve controlling the inlet during the recharging and connecting to the pressure gauge, and a drain valve. The latter serves to empty the receiver of water carried into it by the compressed air.

The blast air receiver is similar to the starting air receiver but of much smaller capacity, as its principal function is to act as a pressure equalizing chamber for the air passing on its way from the final stage cooler to the fuel valves. The head in this case embodies four valves—the inlet valve from the compressor, the outlet valve for the blast air, an "overflow" valve for charging the starting receivers, and a drain valve similar to that on the head of the starting air receiver. There is also embodied a relief valve which serves to protect the receiver itself as well as the final stage cooler and piping of the air compressor.

MECHANICAL DETAILS OF THE TWO-STROKE AIR INJECTION ENGINE

The only parts which differ fundamentally from the corresponding ones on the four-stroke engine are those which have to do with the charging and exhausting of the cylinder. We shall, therefore, refer only to those particular parts of the engine for each of the two main types, namely, the valve-scavenging and the port-scavenging type respectively.

Dealing first with the valve-scavenging type we give in Fig. 33, a sectional drawing which shows its essential features. It will be seen that the cylinder cover is very similar to that of a four-stroke cycle engine, except that there is no exhaust valve. There are, however, two, or sometimes four, air or scavenge valves as well as the usual fuel and starting valves. In the type shown the top portion of the cover is formed into an air chamber to which the scavenge valves are connected by ports. Air is supplied in the usual way by an air pump and is led to the cover by the pipe *A*. The scavenge valves are operated in practically the same manner as the air valves on the four-stroke engine.

The exhaust gases leave the cylinder through the ports *B*, which are spaced completely round the circumference, and are connected by a circular port *C* to the main exhaust pipe *D*.

This type of engine is claimed to have some advantage over the port-scavenging type in regard to the efficiency of the scavenging process, owing to the direct path of the incoming fresh charge from the top to the bottom of the cylinder. In spite of this, however, this type of engine is now receiving little attention, and seems definitely to be disappearing in favour of the port-scavenging type, owing mainly to the greater simplicity of the valve gear and the cylinder cover of the latter.

Turning now to the port-scavenging engine we have in Figs. 34, 35, and 36, illustrations of three well-known types, showing the essential features, i.e. the cylinder, the valves, ports, and piston. Fig. 34 shows the cylinder of the Sulzer type, in which there is a double row of scavenge ports, *A* and *B*, on one side of the cylinder, and a single row of exhaust ports *C* on the other side. The upper row *A* is controlled by a rotary valve, *D*.

During the working stroke the top row of scavenge ports is isolated from the main scavenge air trunk *E* by the rotary valve. The piston, therefore, passes over this row without effect and uncovers the exhaust ports *C* allowing the exhaust gases to flow out, after which the lower scavenge ports are uncovered admitting the scavenge air. At this point the rotary valve *D* opens, connecting the upper row of scavenge ports with the air supply trunk *E*, and this valve then remains open until the piston, on its upward stroke, has covered all three rows of ports. The effect of this is, that even after the exhaust ports are covered again, the cylinder is still in direct communication with the scavenge air supply and, consequently, is positively charged with fresh air up to about the scavenge pressure. This system is an improvement on the original form still largely used on smaller engines of the Semi-Diesel type, in which there is one row of inlet ports only, corresponding to the lower row in Fig. 34, and no method of controlling the time of inlet other than by the piston movement. The advantage of the controlled port system is, of course, that the fresh air has a longer period of entry which extends until after the exhaust ports are closed and thus permits of a certain degree of supercharge depending upon the pressure of the scavenge air supply.

Fig. 35 illustrates the "Mirrlees-Nobel" engine in which very much the same results are obtained, so far as scavenging is concerned, but in another way. In this case the inlet or scavenge ports are of about the same depth as the exhaust ports, but between them and the scavenge air chamber there is a set of light automatic valves, so arranged that they cannot open to allow the scavenge air to enter the cylinder until the gas pressure in the cylinder is slightly lower than that of the scavenge air. By this means the correct timing of the fresh air entry is automatically attained, and at the same time the inward flow continues until the ports are closed by the piston.

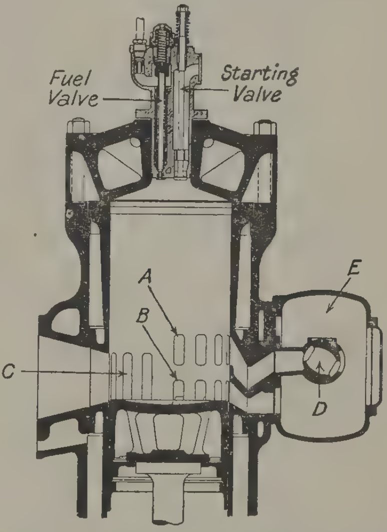

Fig. 34. Cylinder of Two-stroke, Port-scavenging Engine (Sulzer Type)

Very effective scavenging and high volumetric efficiency are claimed for both these systems. It is certain that the cylinder receives a good charge of fresh air at, or rather above, atmospheric pressure and, as a result, high-power output is possible almost equalling that of a normal four-stroke cycle engine.

An important feature of the port-scavenging type is the simplicity of the cylinder cover which contains only two working valves, the fuel valve and the starting valve, of which only the former is in continuous operation. An exceedingly simple structural design is possible, and this is a great advantage as compared with the cover for either a four-stroke engine or a two-stroke valve-scavenging type. The form of the combustion space also is much improved as there are no valve heads or pockets to break the symmetry and smoothness of the internal surface.

The typical opposed piston type is represented in Fig. 36, showing sectional views through one line of a Doxford engine. One view shows the pistons at the end of their outward stroke with the scavenge ports open at the bottom and the exhaust ports open at

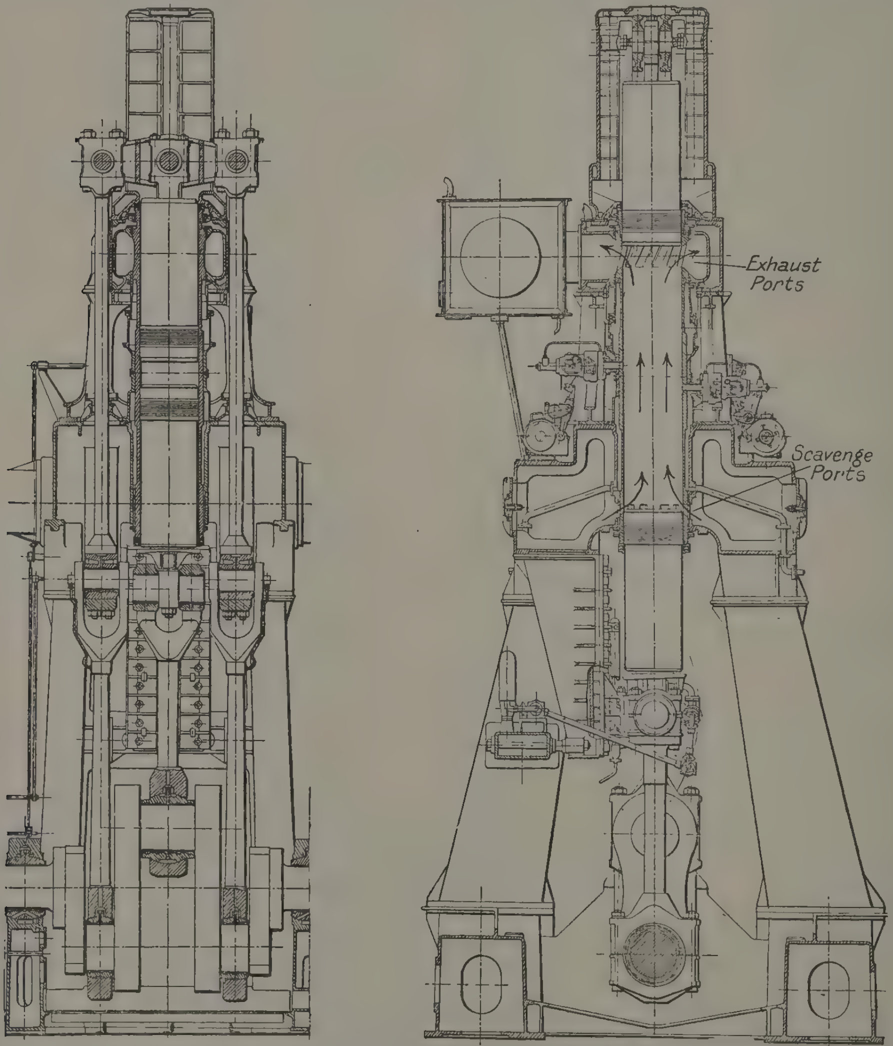

FIG. 36. SECTIONAL VIEW OF 3,000 I.H.P. DOXFORD TWO-STROKE OPPOSED PISTON ENGINE

the top, whilst the other shows the pistons close together at the end of the compression stroke. From this it is seen that as the two pistons on the expansion stroke move away from each other the upper one first uncovers the row of exhaust ports through which the gases begin to flow out, and almost immediately afterwards the lower one opens the air inlet ports admitting the scavenge air which flows in behind the exhaust gases and sweeps the cylinder clear from one end to the other. The ports in both cases extend round the whole circumference of the cylinder, and thus the scavenge action, while being direct and positive, is also equally effective in all parts of the cylinder. The method of driving the upper piston by means of side cranks and rods is also clearly shown in this illustration.

Valve Gear for Two-stroke Engines. Since there are only two valves to operate on the cylinder of a port-scavenging engine, i.e. the fuel valve and the starting valve, the valve gear is exceedingly simple as compared with that of the four-stroke engine. This is a great advantage, particularly for marine engines, as the design and operation of the reversing gear is thereby simplified. Generally speaking also, the absence of any air or exhaust valves results in a much quieter running engine and, of course, the elimination of the exhaust valve, which normally requires more attention than any other on the engine, conduces to reliability of running, especially when heavy or low grade fuel oils are used.

Scavenge Air Pumps. The scavenge pumps usually employed are of the reciprocating type and are incorporated with the engine, the pistons being operated either by links and levers from the main crosshead or directly by separate connecting rods and crossheads from the crankshaft. An example of the former system is seen in Fig. 35, showing those on the "Mirrlees-Nobel" engine, whilst Fig. 37 showing those on the Doxford engine illustrates the latter. A modern development in this connection is the use of a rotary blower instead of a reciprocating pump. In this case, of course, the blower is of the high speed type, being usually directly coupled to an electric motor, and quite independent of the main engine. This system has been adopted to a considerable extent for large marine propulsion engines.

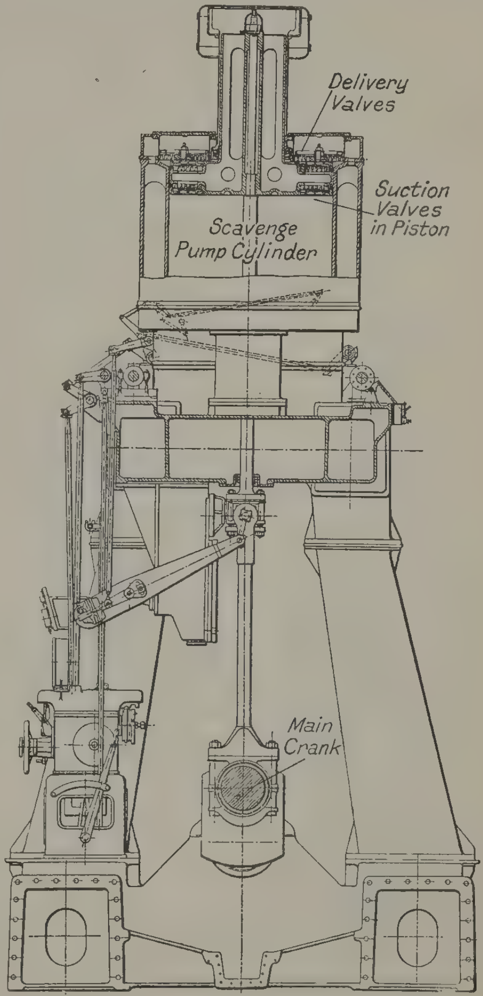

Fig. 37. Section of Doxford Scavenge Pump. Direct-driven Type

THE TESTING OF PRIME MOVERS

By R. Royds, M.Sc., M.I.Mech.E.

LESSON IV

INDICATORS

The pressures in a steam-engine or internal combustion engine cylinder vary too rapidly and through too large a range to be measured by means of an ordinary pressure gauge. It is necessary to have a diagram of pressures so that these pressures may be measured at any point in the stroke or cycle by measuring up the diagram with a suitable scale. The instrument giving such a diagram is usually called an indicator. In the development of his steam-engine, James Watt soon found out the need of an indicator. In his indicator he had a small piston in a cylinder backed by a calibrated spring. The compression of the spring, and therefore the motion of the indicator piston, was proportional to the pressure. He therefore attached a pencil to the indicator piston-rod so that, by touching a paper pinned on a reciprocating board, a diagram of pressures was obtained. The board was given a reciprocating motion from the engine piston-rod and proportional to the motion of that rod or piston. From this diagram he was also able to estimate the horse-power developed in the cylinder, now termed the "indicated" horse-power. This indicator was sufficient for the purpose at the low pressures and speeds then usual. But when a mass is attached to a springy or elastic material, there is a natural tendency for it to oscillate when displaced suddenly, and the time of one oscillation is practically independent of any consideration other than the mass in motion and the strength of the spring. At the slow speeds of rotation in James Watt's time these oscillations of the indicator were rapid enough to obviate any serious error in the diagram, but as pressures and speeds of rotation rose the early engineers following Watt found that these oscillations were very troublesome because, at the higher speeds then usual, the oscillations persisted all round the diagram. It therefore became necessary to speed up the rate of oscillation of the indicator to suit the higher speeds of rotation. They therefore stiffened the spring and reduced the reciprocating mass of the indicator; but with the stiff spring required the height of diagram was very limited, and therefore the next step was to introduce a multiplying link motion of light weight between the indicator piston-rod and the pencil. An obvious modification was to introduce a drum for the paper in place of the board used by Watt. In these brief statements we have the main reasons for the construction of our modern piston and pencil indicators.

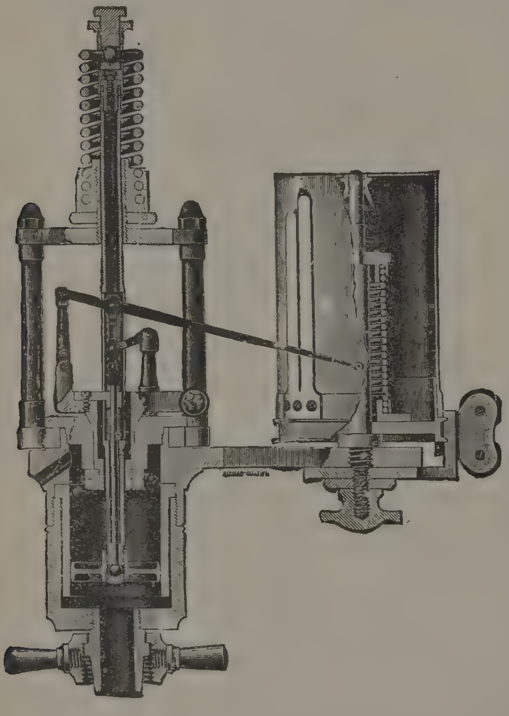

Fig. 17. Crosby Indicator with Outside Spring

There are many makers of this type of indicator, differing only from one another in the details of construction of their indicators.

External Spring Indicators. In the common form of indicator the spring is enclosed behind the piston in the cylinder. The main objections to this arrangement are that the spring is heated when indicating a steam-engine or internal combustion engine, but when used on an air pump, compressor, or water pump the spring is relatively cold. When cold the spring is stronger by about 2 per cent than when used hot. Also any distortion of the spring may cause excessive friction at the piston. In recent years there has been a decided tendency to place the spring outside the indicator cylinder well exposed to the atmosphere, although this arrangement involves an increase in the moving mass. This arrangement is represented in Fig. 17, as built by the Crosby Gauge & Engineering Co., Ltd. The area of the piston is 1 sq. in. for steam-engine indicators, and ½ sq. in. for use with internal combustion engines. This is double the area usually supplied with ordinary indicators, and gives a correspondingly greater active force to compensate for the increased inertia of the longer piston-rod. To reduce friction at the piston a short barrel form is given to the surface of the piston. A small leakage past such a piston is of little consequence as there is a free exit for the steam or gases at the top of the cylinder. The spring can be changed without taking the piston from the cylinder, and the pencil gear is of the usual Crosby pattern. The drum and drum spring are of ordinary construction with the top of the spring fitting on a square on the spindle. By drawing the top of the spring upwards, and turning it round, the tightness of the spring can be adjusted to suit the speed of running so that the drum shall not overrun the indicator cord. A special drum may be introduced if required, instead of the ordinary one, carrying a roll of paper so that continuous diagrams may be obtained when indicating winding engines, rolling mill engines, etc., where the load varies constantly.

Most makers can supply an indicator suited for both steam-engines and internal combustion engines. When used on steam-engines a piston of ½ sq. in. area with a short piston-rod is used, working in the upper part of the cylinder. With internal combustion engines a piston of ¼ sq. in. area is used, having a longer rod so that the piston works in the lower part of the cylinder of corresponding bore.

Indicator Errors. It is no exaggeration to say that many engineers are too liable to accept diagrams taken with ordinary indicators without critical examination or question. No indicator yet built is capable of recording pressures accurately when running at speed, and the best that can be hoped for is that errors may be minimized so as to be practically negligible. The indicator oscillations previously mentioned do not cause serious inaccuracies with engines running at slow speeds of rotation, but if the speed of rotation is too high for the indicator in use these oscillations persist over nearly the whole of the diagram in the manner represented in Fig. 18. It is useless to attempt to get rid of the troublesome oscillations by increasing the frictional resistances in the indicator, such as may be obtained by increasing the pressure of the pencil on the paper. Even a hard blacklead pencil pressing but lightly against the paper may introduce a frictional error of 1 or 2 lb. per sq. in. with a moderate strength of spring, and with the brass or silver pointed pencils marking on chemically treated paper which are commonly used, the frictional error may be much greater. Besides this there is frictional resistance at the indicator piston and pencil gear. As a general statement it may be said that with the piston lightly oiled the piston and pencil gear should fall by its own weight when no spring is in the indicator. This of itself is not always sufficient, as the indicator may behave badly under pressure due to excessive friction. One test for this is to put the indicator under a known steam or gas pressure; then depress the pencil gear and release gently and take a pressure line. After this raise the gear and release gently and draw another line. If there are not more than 2 or 3 lb. per sq. in. difference between these two lines with a moderate pressure and normal strength of spring the indicator is in fair condition in respect to frictional error. A dynamic method of testing for indicator friction under pressure conditions is to depress or raise the piston and release it suddenly, with the pencil in contact with the moving paper on the drum. An oscillating motion is obtained similar to that shown in Fig. 19. The indicator friction is represented by quarter the difference between

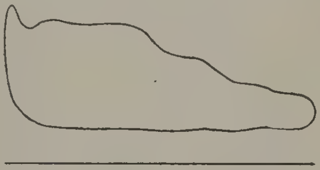

FIG. 18. STEAM-ENGINE INDICATOR DIAGRAM SHOWING EFFECT OF OSCILLATIONS

the heights of *A* and *B* on the diagram. Excessive frictional resistance is liable to cause the indicator to give a stepped form to the diagram as represented in Fig. 20, though it should be recognized that the absence of such steps in a diagram does not of itself signify that the frictional forces are negligible.

The indicator pencil gear usually has about six pins subject to wear, and care should be taken to see that no slackness is allowed to develop at these pins. In many designs those pins most subject to wear are made a conical fit, so that they can be driven farther through by means of a light hammer should wear take place. In any case the pins should be oiled occasionally with light oil, such as sperm oil, of good quality.

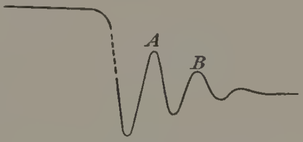

FIG. 19. INDICATOR OSCILLATIONS

Indicator diagrams are usually taken so that the area enclosed by the diagram shall represent to some scale the work done on the engine piston per cycle. For this purpose the drum of the indicator needs to be given a motion which is a copy of that of the engine piston. In operating the drum the string has to be adjusted in length so as to allow the drum to move freely without touching either stop. If the drum is coming in contact with either stop a knock will be felt on touching with the fingers the indicator body near the drum, and to anyone skilled in taking diagrams the knocking will usually be evident from the square corner appearance at one end of the diagram, as represented in Fig. 21, where the missing parts of the diagrams are shown dotted.

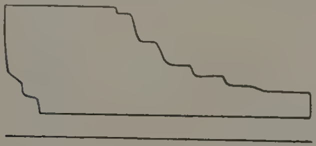

FIG. 20. RESULT OF EXCESSIVE FRICTION IN INDICATOR

Mean Effective Pressure. When the indicated horse-power of an engine is being measured from diagrams the average mean effective pressure is first obtained from the diagrams taken at equal intervals of time. The mean effective pressure is the average height across the diagram, measured as a force according to the scale of the diagram. It can be shown that the work done on the engine piston per unit of piston area per cycle is given by M.E.P. × stroke of engine, and, as the M.E.P. is independent of the length of the diagram provided the motion of the drum is a copy of that of the engine piston, it is usual to operate the indicator drum from the engine piston-rod or crosshead through some form of indicator reducing gear.

Indicator Reducing Gears. Three arrangements of indicator reducing gears suitable for short stroke engines are given in Fig. 22, (*a*), (*b*), and (*c*). That shown as (*c*) is a modification of the other two suitable for single-acting gas engines. In all three cases a bracket supported from the engine frame accommodates the pin *B*, about which the arm *AB* swings. In (*a*) a pin *A* on the crosshead moves through the stroke A_1A_2, and operates the arm through the slot shown. The indicator string is operated from the stud *E* on the arm. This arrangement does not give to *E* quite a true copy of the motion of *A*, but unless *AB* is very short the discrepancy is almost negligible. By placing the slot on the crosshead and the pin *A* on the arm, as shown at (*b*), the gear is theoretically correct. In all these cases the indicator string *I* should be parallel to the line of stroke of the engine.

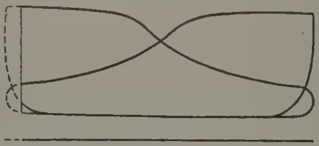

FIG. 21. INDICATOR DRUM TOUCHING STOP

For long stroke engines the pantograph type of gear shown in Fig. 23 forms a compact arrangement, while for vertical engines a modification of Fig. 22 (*c*) is commonly used. These various gears are not usually convenient for completely enclosed engines. For such cases a small eccentric on the crank shaft, set in line with the engine crank, is a suitable arrangement provided the ratio between the length of the eccentric rod and the throw of the eccentric is the same as that between the engine connecting rod and crank.

The indicator cord should be made as short as possible to avoid excessive stretching when in operation. Indicator cord in which strands of copper wire are incorporated is to be preferred to plain cord. If the cord is of considerable length, as might be the case with long stroke engines and tandem engines, it is better to connect a piece of Bowden wire to the indicator gear and keep it taut by means of a spring at the far end. Clips on the wire would be used to which the indicator string of short length would be connected when indicating the engine.

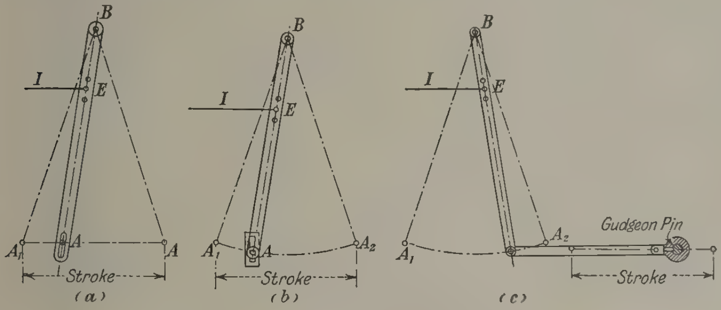

FIG. 22. INDICATOR REDUCING GEARS

Indicator Cocks. The connection between the engine cylinder and the indicator should be as short as possible to avoid excessive lag of pressure. For this reason it is preferable to screw the indicator directly into the indicator hole in the cylinder. This connection is commonly screwed ¾ in. Whitworth, but, in the writer's opinion, this gives too small a hole with a brass cock, and the connection and the hole in the cock could be made larger with advantage. In double acting steam-engines a common, but faulty, arrangement is to bring a pipe, say, ⅝ in. or ¾ in. bore, from both ends of the cylinder to a three-way cock, to which the indicator cock, or in some cases the indicator itself, may be attached. By this means one indicator serves to take cards from both ends, but the length of piping and the number of turns are objectionable. Before taking a card the pipes should be "blown" through to clear out any water which may be lying there. When taking an atmospheric line the hole in the cock should place the indicator piston in direct communication with the atmosphere.

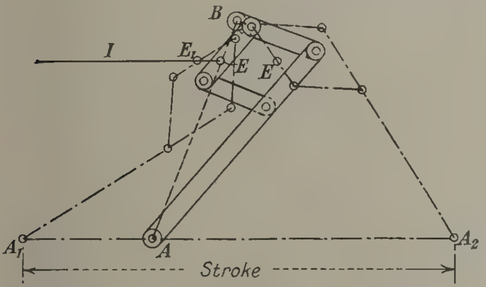

FIG. 23. PANTOGRAPH TYPE OF INDICATOR GEAR

Calculation of Indicated Horse-power. In determining the indicated horse-power of a reciprocating engine, compressor, or pump, the average mean effective pressure has to be obtained from the indicator diagrams. With reference to the end of the cylinder from which diagrams were taken, the mean effective pressure may be defined as the average pressure acting on the piston at that end during one cycle, taking into account that on the forward stroke the pressure is positive and on the return stroke it is negative. Thus, in effect, it is the average height across the diagrams expressed in terms of pressure according to the scale of the diagram, measurements being made at right angles to the atmospheric line. Two methods are in common use for measuring M.E.P., viz., the mid-ordinate method and the planimeter method. The mid-ordinate method is shown in Fig. 24 applied to a steam-engine diagram, where *AL* is the atmospheric line. The two lines *GH* and *IK* are first drawn at right angles

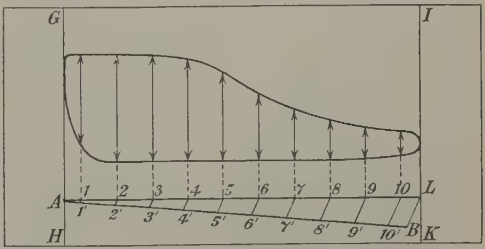

FIG. 24. MID-ORDINATES FOR MEASURING M.E.P.

to the atmospheric line touching the extremities of the diagram. If a line *AB* is drawn, say, 5 in. long, divisions 1^1, 2^1, 3^1, etc., would be marked off on the line at $\frac{1}{4}$, $\frac{3}{4}$, $1\frac{1}{4}$ in., etc., along *AB*. Joining *B* to *L* and drawing parallels as shown gives the points 1, 2, 3, etc., for the 10 mid-ordinate positions. With the appropriate scale, the distances shown by the lines with arrow-heads would be measured and tabulated on the card over the respective ordinates. Adding these together and dividing by 10 gives the mean effective pressure for that diagram. All the cards for each end of the cylinder and for all cylinders would be treated similarly, and the average value for each end of each cylinder then obtained. If found to be convenient it is usually quicker to place the 5 in. line *AB* so that the end *A* lies on *GH* and the end *B* on the line *IK*. In that case the parallels $1^1 1$, $2^1 2$, etc., are not required.

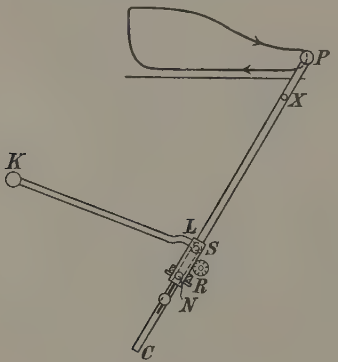

Fig. 25. Planimeter

The planimeter method is quicker and, if carefully done, is probably more accurate than the mid-ordinate method. The ordinary planimeter is shown in Fig. 25. It consists of a link *KL* pivoted at *L* to the slide *S*, which can be shifted to any position along the bar *CP*. On the slide is a roller *R* carrying a scale, and when in use the edge of this roller runs on the fixed paper underneath it. The purpose is first to measure the area of the diagram. The card is first pinned on to a smooth board and the planimeter is then set in a position similar to that shown in Fig. 25, with the pin at *K* fixed securely in the board. The tracing point *P* is then placed at any convenient point on the diagram. The roller *R* rests on a piece of plain paper pinned tightly on the board, and the reading on the scale is observed. The point *P* is now carried clockwise round the diagram until the starting-point is reached and the reading on the scale again observed. The difference between the two readings represents the area of the diagram. The slider *S* is previously set at a definite point on the rod *CP*, according to the units required in measuring areas. The M.E.P. is then given by dividing the area by the length of the diagram and multiplying by the "strength" of the indicator spring. Thus, if the area is 2·1 sq. in., and the length of diagram 3·7 in. with a 60 spring, then,

$$\text{M.E.P.} = \frac{2{\cdot}1}{3{\cdot}7} \times 60 = 34{\cdot}1 \text{ lb. per sq. in. nearly.}$$

The mean effective pressure may be obtained more directly by adjusting the length between the two points *X* and *N* to the length of the diagram. The difference between the two readings divided by a constant for the instrument gives the mean effective pressure directly. The constant can be found by testing the planimeter on a rectangle of known height.

When calculating the indicated horse-power the best procedure is to calculate for each end of each cylinder separately and then add the results together to get the total i.h.p.

Let P_m = mean effective pressure, lb. sq. in.
,, A = area of piston, sq. in.
,, L = length of stroke of engine, ft.
,, N = number of cycles per minute.

The work done at the one end of the piston per cycle is $P_m \times A \times L$ ft.-lb. Therefore—

$$\text{i.h.p.} = \frac{P_m \times A \times L \times N}{33{,}000}$$

Example. Cylinder diameter, 11·5 in.; piston-rod diameter (bottom end of cyl.), 3 in.; stroke, 36 in.; revolutions per minute, 95. Values of P_m are 68 and 64 lb. per sq. in. at top and bottom ends respectively.

Top.

$$\text{i.h.p.} = \frac{68 \times {\cdot}785 \times 11{\cdot}5^2 \times \frac{36}{12} \times 95}{33{,}000}$$
$$= 61 \text{ nearly.}$$

Bottom.

$$\text{i.h.p.} = \frac{64 \times {\cdot}785 \times (11{\cdot}5^2 - 3^2) \times \frac{36}{12} \times 95}{33{,}000}$$
$$= 53 \text{ nearly.}$$

Total i.h.p. = 61 + 53 = 114

REFRIGERATING MACHINERY

By G. W. Daniels, M.Eng., Wh.Ex., A.M.I.Mech.E.

LESSON II

MACHINES

For many years past the ammonia refrigerating machine had developed into a more or less standard type of machine, usually horizontal and always slow running, as illustrated by Fig. 2. This shows a section through the compressor cylinder. It consists of a cast-iron barrel with a cast-iron liner. Spherical shaped

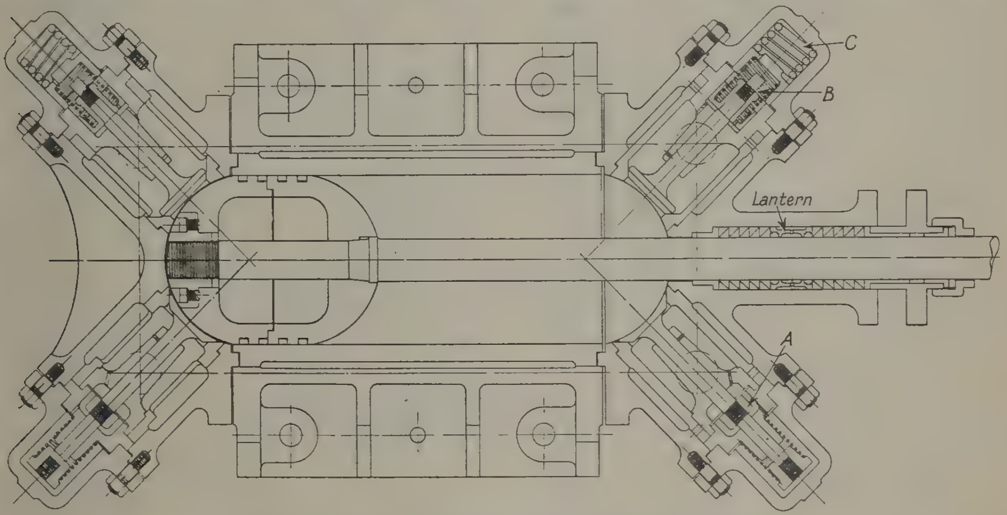

Fig. 2. Section of Slow-speed Ammonia Compressor

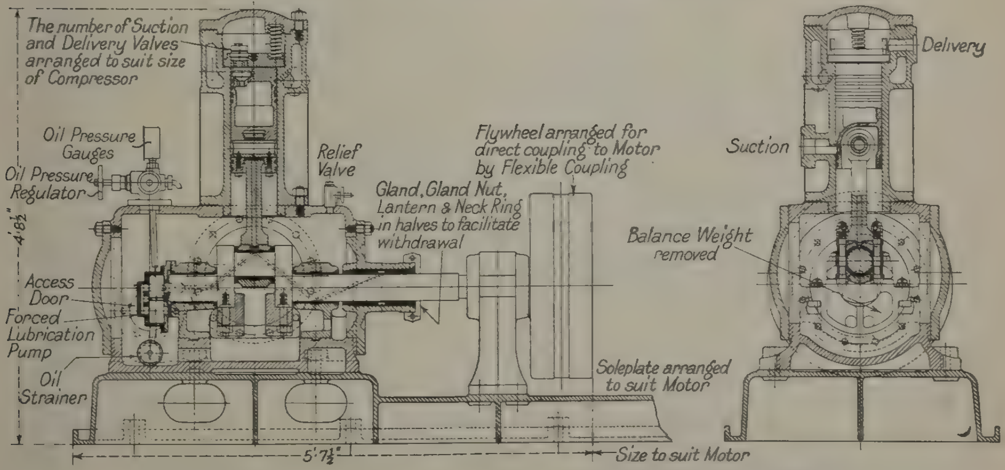
The number of Suction and Delivery Valves arranged to suit size of Compressor
Oil Pressure Gauges
Oil Pressure Regulator
Relief Valve
Gland, Gland Nut, Lantern & Neck Ring in halves to facilitate withdrawal
Flywheel arranged for direct coupling to Motor by Flexible Coupling
4'8½"
Access Door
Forced Lubrication Pump
Oil Strainer
Soleplate arranged to suit Motor
5'7½"
Size to suit Motor
Delivery
Suction
Balance Weight removed

Fig. 3. Section of High-speed Single-acting Ammonia Compressor

covers are fitted at either end which carry the suction and delivery valves. The object of the spherical covers is to allow of valves of the largest possible area being fitted without making any pockets or other clearance spaces in which compressed gas could accumulate. Such clearance spaces have a very adverse effect on the working of the machine and are kept as small as possible. For the same reason the compressor piston has rounded ends to fit snugly into the covers. The distance between the cover and piston at the end of the stroke is only $\frac{1}{32}$ in. The suction valves open into the cylinder, and are fitted with nuts *A* on the middle of their spindles which prevent the valves falling into the cylinder if the spindles break. The delivery valves are fitted with two springs, *C* and *B*. In normal working only *B* is brought into use, but if extra opening is required at any time, the spring *C* automatically comes into operation and permits the delivery valves to have extra lift and opening. The packing in the stuffing box is divided into two parts by a ring, called a lantern. The space inside the lantern is connected to the compressor suction pipe and, therefore, if any ammonia leaks past the first set of packing it is drawn off into the suction, and the outer set of packing has only to hold tight against the comparatively low pressure existing on the suction side.

FIG. 4. RING PLATE VALVE FOR HIGH-SPEED COMPRESSOR

During the compression of the refrigerant in the compressor, heat is generated and we have to keep the compressor from becoming overheated. This may be done in either of two ways, (*a*) by fitting the compressor with a jacket through which water circulates, or (*b*) by allowing a small portion of liquid refrigerant to enter the compressor along with the vapour from the refrigerator. The small amount of liquid evaporates during the compression, absorbing the heat generated and keeping the compressor cool. This latter method of working

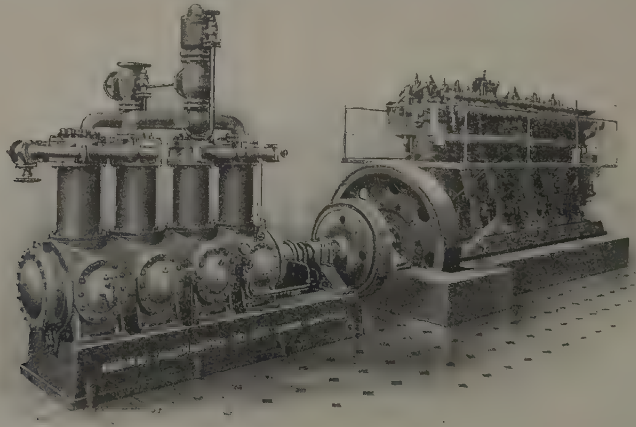

FIG. 5. HIGH-SPEED AMMONIA COMPRESSOR
(Haslam Engineering Co.)

is usually called "wet compression." The first method, where no liquid refrigerant is admitted to the compressor but a water jacket used instead, is called "dry compression."

Machines of this type gave very good service, but owing to their low speeds, from 45 to 130 r.p.m., they occupied large floor space and were costly.

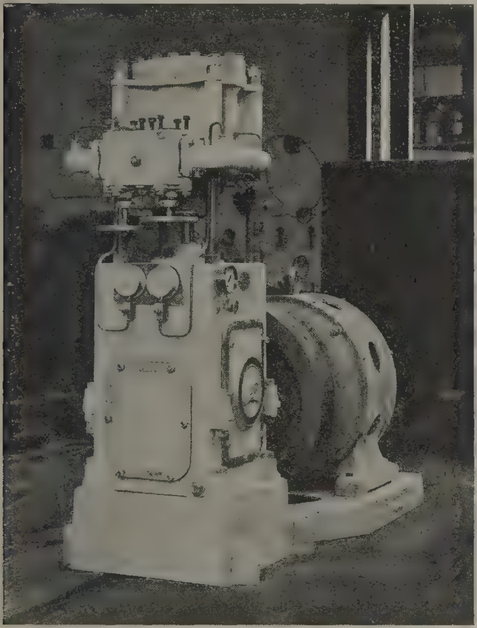

FIG. 6. HIGH-SPEED DOUBLE-ACTING AMMONIA COMPRESSOR

The large space and high first cost of the slow-speed refrigerating machines gradually brought about a demand for quicker running and less costly machines.

Fig 3 shows a section through an ammonia machine of this type. This machine is single-acting and has no piston-rod. The piston is of the trunk type and is directly attached to the connecting rod. The cylinder is mounted on a strong crankcase containing all the moving parts which are entirely enclosed. An oil pump supplies lubricating oil under pressure to all bearings, and also seals the stuffing box through which the crankshaft passes out of the crankcase. The suction valves are fixed in the top of the piston, while the delivery valves are carried in a false head, closing the working barrel of the cylinder, and held down by springs. Valves in these machines are of very light weight, so that they can open and close quickly. Fig. 4 illustrates a valve of this kind which is known as a ring or plate valve. Machines of this description run at speeds up to 450 r.p.m., and are made with 1, 2, 3, or 4 cylinders mounted on one crankcase.

Fig. 5 shows an external view of a high-speed single-acting ammonia machine which is generally similar to the one just described, with the exception that in this case the suction valves are carried in the cylinder head and not in the piston.

Fig. 6 shows a double-acting, high-speed ammonia refrigerating machine. In this case there is a piston-rod passing through a stuffing box. Such machines occupy very small space, and are comparatively cheap for the amount of refrigeration produced. If properly designed, they are very satisfactory.

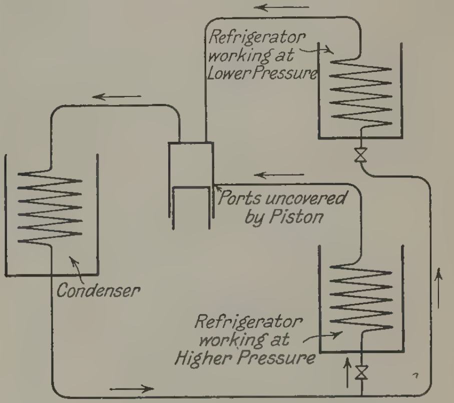
Refrigerator working at Lower Pressure
Ports uncovered by Piston
Condenser
Refrigerator working at Higher Pressure

FIG. 7. DIAGRAM OF TWO-PRESSURE SYSTEM OF WORKING

Sometimes the compressor is arranged to take the vapour from two separate refrigerators working at different pressures. The vapour at the lower pressure is drawn into the cylinder in the usual way through the suction valves. When the piston reaches the end of its stroke it uncovers openings or ports in the cylinder, and through these ports the vapour from the second evaporator rushes in by virtue of its higher pressure. We are thus enabled to make one compressor cylinder take in two charges of refrigerant at each stroke instead of one and, consequently, we get a much larger amount of refrigeration from the machine.

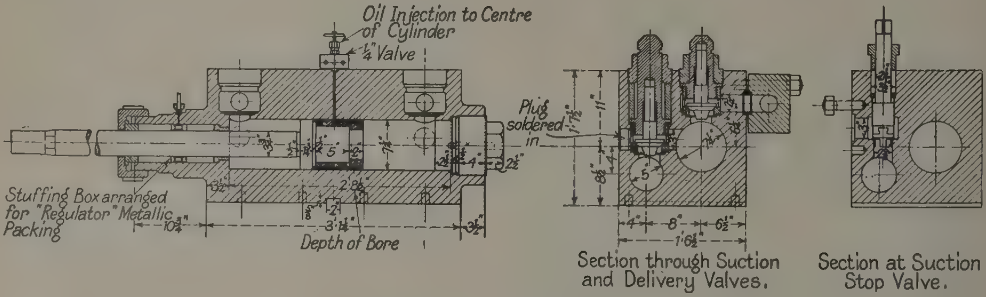
Oil Injection to Centre of Cylinder
¼" Valve
Plug soldered in
Stuffing Box arranged for "Regulator" Metallic Packing
10¾"
3' 1¼"
3½"
2' 8½"
Depth of Bore
5"
7¼"
4"
2½"
11"
1' 7½"
8½"
4"
8"
6½"
1' 6½"
Section through Suction and Delivery Valves.
Section at Suction Stop Valve.

FIG. 8. SECTION OF CO_2 COMPRESSOR

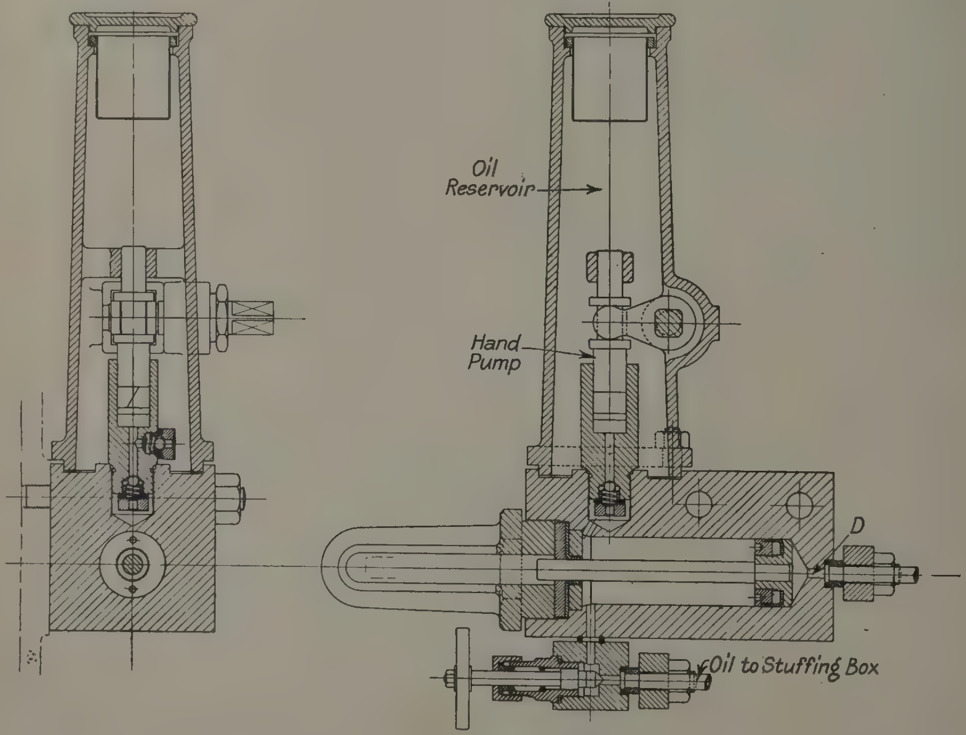

FIG. 9. PRESSURE LUBRICATOR

Fig. 7 shows, diagrammatically, a compressor cylinder working in this way. This system should be used on all suitable occasions.

SO_2 machines have followed very closely the general arrangement of the slow-speed horizontal ammonia machine illustrated in Fig. 2, and do not need further description.

With regard to CO_2 machines, Fig. 8 illustrates a typical CO_2 compressor cylinder in section. This is made from a solid forging of high grade steel, in order to withstand the high pressure to which it is subjected.

On no account should cast iron be used for CO_2 compressors, although small sizes have had their cylinders made from gun-metal. The chambers containing the valves are bored out in the solid steel block, and the valves, seats, and guides are made of a high grade nickel steel. The compressor pistons are fitted with piston rings of cast iron, or sometimes with leathers after the manner of the hydraulic ram. The stuffing box is again divided into two parts by a lantern as in the case of the ammonia machine, but instead of connecting this space to the compressor suction, we pump oil into the space under a pressure greater than that of the CO_2. Hence any leakage that takes place is of oil into the system and not of CO_2 out of the system. Fig. 9 shows the type of simple oil pump used for this purpose, and it is usually called a pressure lubricator. It consists essentially of a steel block bored out to form a working barrel and fitted with a piston and rod. On top of the block is placed an oil reservoir in which is situated a hand-operated oil pump. By working the pump, oil is delivered into the cylinder and forces the piston to the right. CO_2 at the highest pressure in the system is then admitted at *D* on the other side of the piston, and this pressure gradually forces the piston to the left and so delivers the oil into the stuffing box. The two sides of the piston are of different area owing to the presence of the piston-rod. The CO_2 always acts on the larger area of the piston and, hence, the oil acting on the smaller area of the piston, must automatically become at a greater pressure than the CO_2. The following ratio holds good—

$$\frac{\text{oil pressure}}{CO_2 \text{ pressure}} = \frac{\text{larger area of piston}}{\text{smaller area of piston}}$$

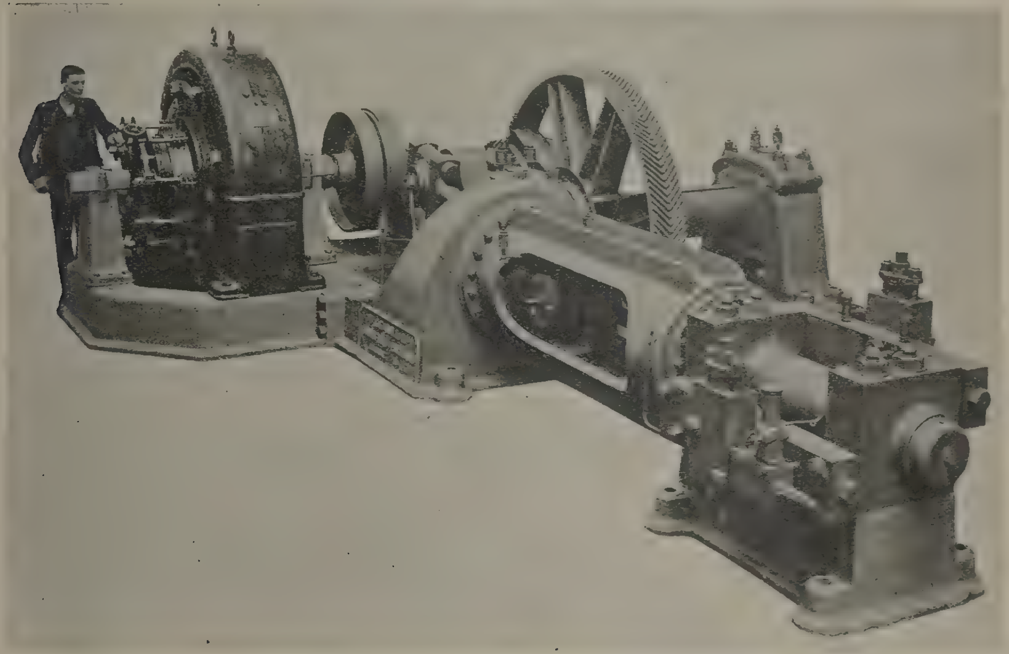

Fig. 10. Slow-speed Gear-driven CO_2 Machine

The whole forms a very simple and effective device without any moving parts in continuous operation.

The general arrangement of a CO_2 machine may take many forms, horizontal, vertical, etc., just like the ammonia machine. In fact, all the moving parts, connecting rod, etc., can be the same for either type of machine, only the compressor cylinder and details being of special design in each case.

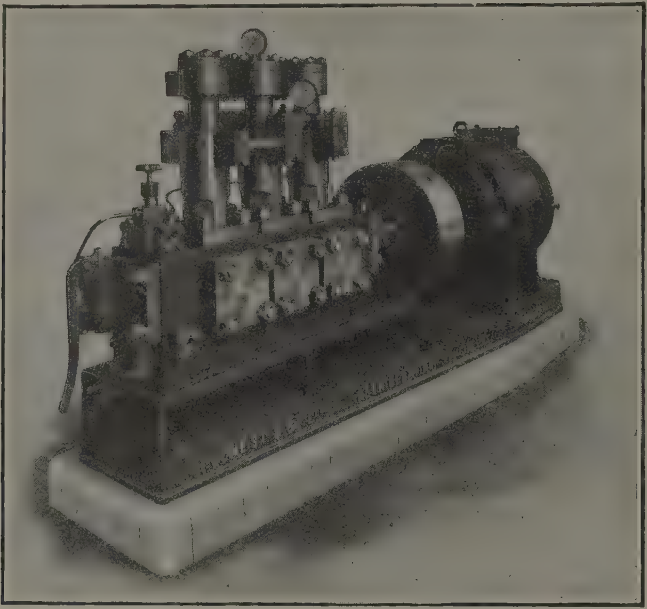

Fig. 11. High-speed CO_2 Machine

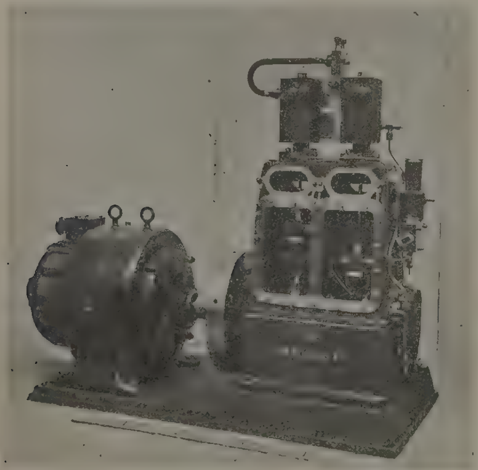

Fig. 12A. High-speed CO_2 Machine

Fig. 10 shows a horizontal gear-driven slow-speed CO_2 machine.

In common with ammonia machines, types of CO_2 machines running at a high speed have also been developed. Fig. 11 shows an interesting example of such a machine. This is a single-acting compressor, having trunk pistons directly attached to the connecting rod and with the suction valves carried in the piston. The delivery valves are mounted on a false head just as in the case of the ammonia machine. The crankcase is made from a solid steel forging, as it may be subjected to a very high pressure if there is any leakage of CO_2 past the piston rings. The crankshaft passes through an oil-sealed stuffing box, and an oil pump provides forced lubrication to all moving parts. The cylinders are also machined from solid steel forgings.

In contrast to Fig. 11, Figs. 12 and 12A show another type of high-speed CO_2 machine, also single-acting. In this case, however, the cylinders are fitted with pistons and rods in the ordinary way, the piston-rods passing through a stuffing

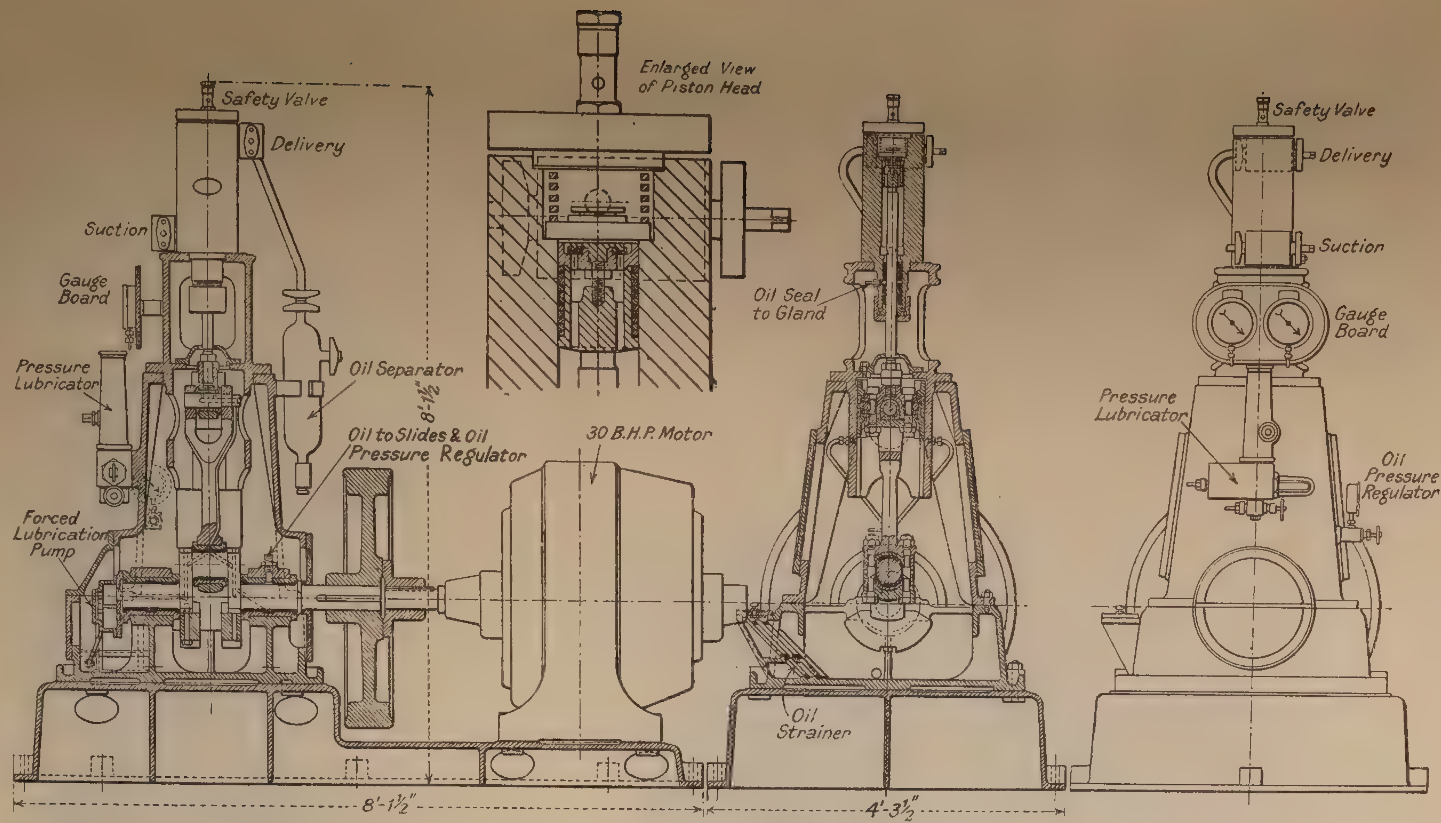

FIG. 12. HIGH SPEED CO_2 MACHINE

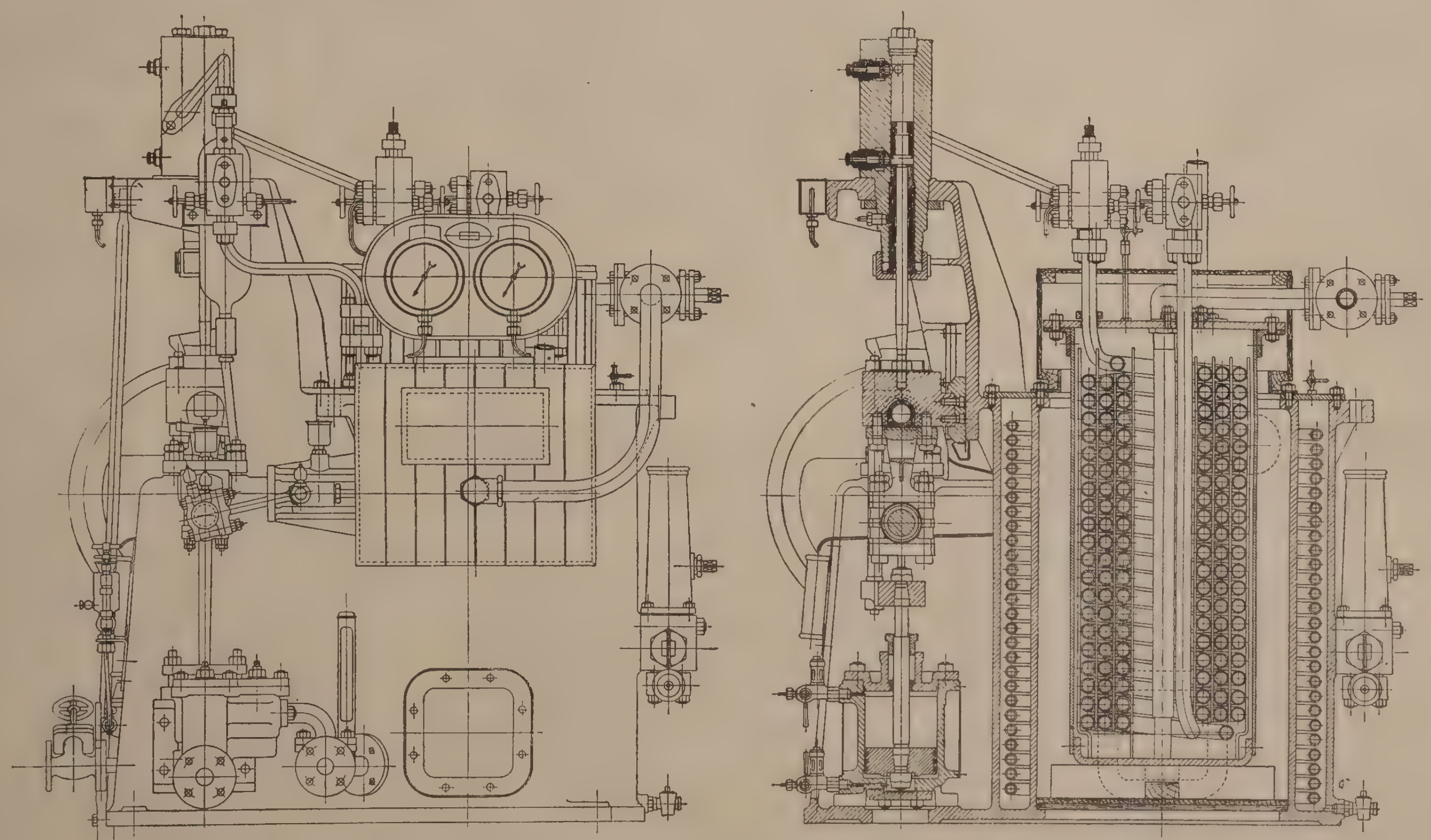

FIG. 13. STEAM-DRIVEN CO_2 MACHINE FOR SHIP'S PROVISIONS

box and being connected to a crosshead. All the motion work, connecting rod, etc., is totally enclosed in a light cast-iron crank-case. Forced lubrication is provided to all bearings. The suction valves are mounted in the piston and the delivery valves in a false head.

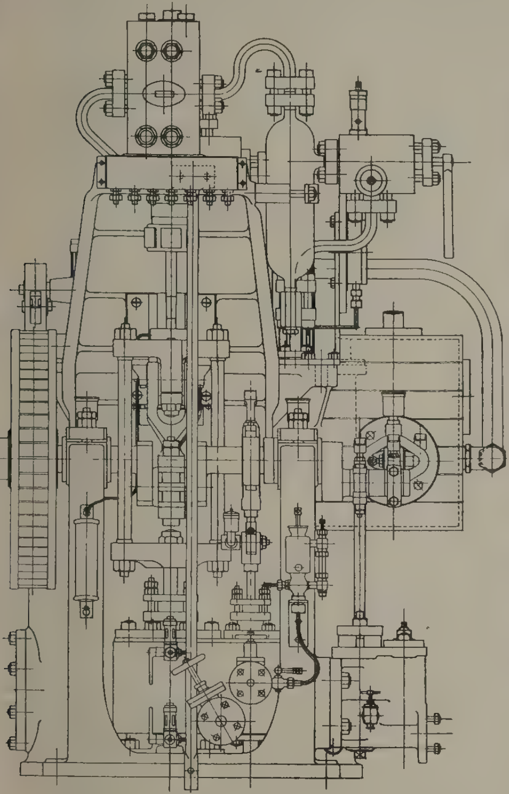
FIG. 13A. STEAM DRIVEN CO_2 MACHINE FOR SHIPS PROVISIONS

High-speed machines have not as yet made any appreciable headway in marine refrigerating work. Here the use of CO_2 is almost universal, and Figs. 13 and 13A show a type of machine used for the preservation of the ship's provisions on cargo boats. The machine consists of a double-acting vertical CO_2 compressor mounted on a cast-iron baseplate. The cylinder of the steam-engine driving the machine is placed underneath the compressor cylinder, the crankshaft coming between them. The respective pistons are coupled to the crosshead, as shown in Fig. 13. The condensing coil is situated in an annular space formed in the baseplate casting, while in a larger interior space in this casting there is placed a mild steel tank containing the evaporator coils. This tank is suitably insulated with granulated cork. A water pump for supplying

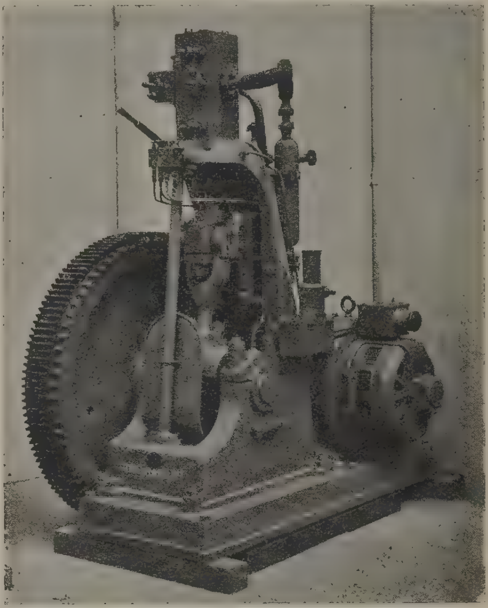
FIG. 14. HALL MOTOR-DRIVEN CO_2 MACHINE

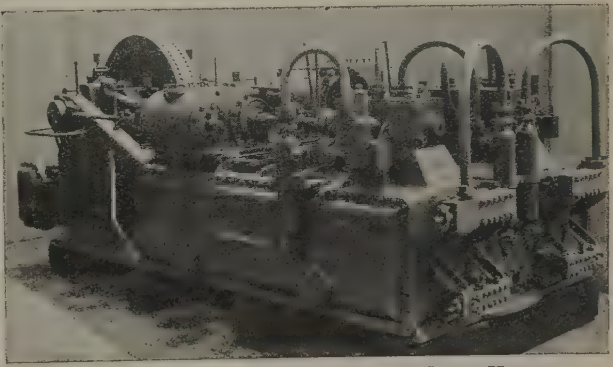
FIG. 15. MACHINE FOR REFRIGERATING CARGO HOLDS
(J. & E. Hall)

the condensing water and a brine pump for circulating the cold brine are driven off the crankshaft. The whole plant is thus self-contained, and is a useful and inexpensive unit for the work intended.

Fig. 14 shows another type of CO_2 machine for ships' provision chambers, in this case driven by an electric motor through gearing.

Fig. 15 shows a large CO_2 machine for refrigerating the cargo holds. This is of the horizontal type, having two compressor cylinders placed tandem with the cylinders of the compound steam-engine which drives them. The crankshaft of such a machine is in two parts, coupled at the middle, and the steam piping is so arranged that either side of the machine can work quite independently of the other. In such a case, the low pressure steam cylinder would be supplied with steam direct from the boilers. Such a machine is described as "duplex," on account of its two compressors being capable of independent operation, and it is almost universally adopted for marine work on account of its giving the greatest protection against total stoppage in the event of failure of any part of the machine. The CO_2 condenser coils are arranged in the hollow bedplate of the machine. In some cases a steam condenser is also arranged in the bed at the front end under the engine. When this is done, an air pump and a feed or hotwell pump is often driven off the crankshaft.

SOLDERING, BRAZING, AND WELDING

By E. A. Atkins, M.I.Mech.E., M.I.W.E.

LESSON I

The jointing of metals is one of the most important functions with which the engineer has to deal, as a little consideration will show what an enormous amount of practical work of one kind and another resolves itself into the correct forming of joints. Therefore, every engineer should have a working acquaintance with the various ways by which metals may be connected together.

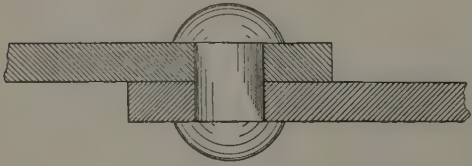

Fig. 1. Single Riveted Joint

Before considering the processes of soldering. brazing and welding, it will, perhaps, not be out of place to first of all make a comparison of the general methods that are in common use for the uniting of metals. These may be summarized as follows—

(*a*) Riveting and bolting.
(*b*) Grooving.
(*c*) Soldering.
(*d*) Brazing.
(*e*) Welding.

Riveted Joints. Of all the methods that are in use for jointing metals, it is probable that riveting is the oldest. Whilst it gives a very secure joint, unfortunately, where articles are required to stand great pressure this kind of connection in its simple form is not very efficient. The percentage strength of the best form of riveted joint depends upon several factors, such as whether or not the plates and rivets are of steel or iron, or whether the holes are punched or drilled. Generally speaking, a joint that is formed of drilled holes is stronger than a joint of which the holes have been punched out of the plate. In addition to the extra strength obtained by the drilled holes, there is also greater uniformity in the quality of the joint and less danger of the plates being locally strained through holes having to be drifted on account of want of alignment.

A single riveted joint, such as shown in Fig. 1, is, of course, the weakest form of joint that can be constructed, and has an efficiency of about 55 per cent; whilst a double riveted joint would have an efficiency of about 70 per cent, a treble riveted joint attaining an efficiency of about 80 per cent.

From what has been said it will be seen that there is considerable waste of material through having to use riveted joints. This is specially so in the case of a single riveted joint, for where a vessel has to be constructed to withstand a certain pressure, it is inevitable that it must be designed for the joint to be sufficiently strong to meet the conditions required. Where a single riveted lap joint is used, on account of its strength being only about half that of the solid plate, it results in plates of almost twice the required thickness having to be used.

Several devices, such as thickening up the edge of the plates which cover the riveted joint portion, have been tried so as to overcome the inherent weakness of this kind of joint; but these have all been too costly to put into general use and have been abandoned.

Grooved Joints. It is not generally realized what a valuable method of jointing is given by grooving (Fig. 2), and there is not the least doubt that the inventor, whoever he may be, of this method of connecting the edges of sheet metal has conferred a very great boon upon the metal worker. If properly made this kind of joint can even be watertight without the aid of solder, but if made in tin-plate or other metal that can be soldered it represents one of the very few joints that is as strong as the sheet from which it is formed.

Whilst a simple form of grooved joint is shown in Fig. 2, it can readily be realized that this method of connecting the edges of sheet metal may take many forms. For instance, in addition to being used for the side seams of articles, it can be applied for fastening bottom and side plates together, corner joints, transverse joints for pipes, etc. Where extra strength is required, it is sometimes used to make a joint which is afterwards to be soldered; and in some special cases is actually used in sheet metal which is to be welded, the four thicknesses of the metal simply being melted down by the use of the oxy-acetylene blow-pipe.

Soldered Joints. In these days of wireless telephony, mechanics of every possible kind and description succeed in using a copper bit to form a simple soldered joint in wire, sheet or plate; and whilst the operation of tacking together two bits of tinned wire is not very difficult, it is not such an easy matter to form a good solid joint where comparatively large surfaces have to be connected. For successful soldering it is necessary to take notice of a few important points—

The solder should be of such a composition as to be suitable for the kind of work required to be done. For good all-round work this is usually composed of one part of tin to one part of lead. For commoner work two parts of tin to three parts of lead is sometimes used. For higher class work two parts of tin to one part of lead.

The composition of the solder, however, can be varied to suit any special circumstances. Occasionally a small amount of antimony is added to the solder, but on no account should zinc be present in the solder, as this tends to make it run very pasty and form a very coarse joint.

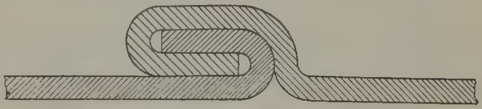

FIG. 2. SECTION OF GROOVED JOINT

If a solder is required of a particularly low melting point, then the metal bismuth is added to the lead and tin in a proportion of about two parts of bismuth to one part of lead and one part of tin.

The fluxes or materials which are used to assist the flow of metals are various, but the one most commonly used is known as "killed spirits," this being nothing more nor less than hydrochloric acid which is converted into a solution of chloride of zinc by adding zinc to the acid until it is completely "killed."

A solution of chloride of zinc is an excellent all-round flux, and can be used with all metals with the exception, perhaps, of aluminium. Its chief disadvantage is that it tends to set up corrosion at the joint unless it has been completely washed away or neutralized by using water in which a little common soda has been dissolved.

On account of the disadvantage named, the greatest care should be exercised in using it in connection with electrical work. Indeed, it is generally the best plan for this purpose to use a flux which will not set up subsequent corrosion. Other fluxes used are resin, salammoniac, and several paste and liquid fluxes which are put on the market.

The mistake which the inexperienced usually make in soldering a joint is to stick the metal on like glue or putty, instead of holding the soldering-iron long enough against the joint for the solder to be properly melted so that the solder and metal will firmly adhere together. Instead of using the extreme point of the soldering iron to run the solder on the joint, an edge of the square point should be used to draw along the solder. In this way a greater quantity of heat will be transmitted to the joint, and thus a better and quicker joint made. It should be remembered that the most important function of the soldering-iron is to give out its heat to the solder and to the joint, thus to facilitate a proper combination.

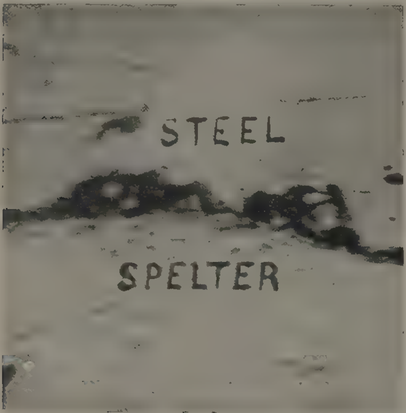

Fig. 3. Photomicrograph of Section of Brazed Joint to a Magnification of 100 Diameters. The Black Layer Shows the Copper-iron Joint

Care must be taken that the soldering iron does not get red-hot, or else the tinning on its point will be burnt off, or, what is worse, form a hard skin of bronze which is somewhat difficult to file away. When the soldering-iron is drawn from the fire it can be cleaned by quickly dipping the point into the spirits, and also in this way after a little practice one can judge as to its proper temperature. If, when dipping the iron into the spirits, much smoke is given off or the liquid spurts about, the soldering bit is too hot, or if on the other hand small bubbles of spirit adhere to the tinning on the copper, it will be found that it is not hot enough.

After a little practice it will not be at all difficult to observe the proper temperature at which to use the soldering-iron.

If the iron does become overheated, and the tin on its surface becomes burnt hard, then it should be carefully filed at the point to remove all the bronze, and when it has fallen to the necessary temperature then tin same by dipping in the "killed spirits," and applying a little solder to the four surfaces just about the point.

In soldering zinc or galvanized iron care should be taken not to get the soldering bit too hot, as some of the zinc will be melted from the surface of the sheet and will mix with the solder, making the latter very rough.

In soldering tarnished zinc or galvanized iron the spirits should not be quite "dead," i.e. the scrap zinc should be withdrawn from the acid before the boiling action has quite ceased. It is, perhaps, a better plan to freshen up the "killed spirits" by the addition of a small quantity of neat acid.

In soldering copper, brass, and black iron the edges of the metal should be carefully cleaned before being placed together, and, as a matter of fact, before they are brought into contact these should be carefully covered over with a thin film of the flux, as it should not be forgotten that the molten solder will not run on any surface that has not previously been treated with a flux. One of the tests of a good soldered joint is that the solder should have run right through same, and if this is done a really good solid joint will be the result.

Where surfaces have to be tinned for setting together, care should be taken to see that they are perfectly clean, then covered with a flux and tinned. However, before the surfaces are again brought together a slight amount of flux should be applied to them, and after placing in contact, the joint heated by the application of a soldering-iron, blow-lamp, etc., when a good solid connection should result.

Brazed Joints. Every working engineer, in addition to being able to make a soldered joint, ought to learn the art of brazing, as it can be quite easily acquired and applied by the metal worker to many simple kinds of jobs.

Brazing is somewhat similar to soldering, except that a much harder solder, commonly called brazing spelter, is used, and it is necessary to apply a much higher temperature to form the joint than in the case of soft soldering.

When a brazed joint is properly made it is exceptionally strong, in many cases being equal in strength to the metal which is brazed together. Its chief disadvantages are that it is somewhat stiffer and harder than the base metal, consequently, it has a greater tendency to fracture under bending.

The great strength of a brazed joint, as compared to a soldered joint, is due to the fact that the spelter or brass used actually combines with the iron or steel to form a distinct compound, which firmly combines the spelter with the steel. This layer between the brass and the steel is really an alloy of copper and iron, and can be quite distinctly seen in the photomicrograph, Fig. 3. As this photomicrograph showing the cross-section of a brazed joint is to 100 diameters magnification, it will readily be understood that the connecting layer is very thin.

Brazing methods can be applied to iron, steel, copper, and brass, if suitable spelters and fluxes are used for the particular metals.

Several methods are in use for forming brazed joints. For band-saws, thick plate, and other kinds of articles, the two edges of the metal are usually thinned down by filing or grinding in the form of a wedge, and in brazing are clamped in such a way as to allow the spelter to run right through the joint.

For thin sheet metal the method shown in Fig. 4 is adopted, in which the edge of one sheet, in this case marked *A*, is just nicked or cut in, and the clamps as they are called, marked *C*, simply lifted up. The edge of the other sheet, marked *B*, which is not cut at all, is then slipped between the clamps, these being carefully hammered or closed down on to the sheet *B*.

The flux commonly used is powdered borax, which is made up into a paste; and when the brazing spelter and flux are to be applied together, the borax paste and the brazing spelter in a fine state of division are carefully mixed together. For good brazing, however, it is always the best plan to apply a thin solution of borax water to the joint and let this run right through. It may be taken as an axiom in brazing that the molten brass or spelter will only follow through those parts of the joint that have been first coated with a fluxing solution.

If the spelter is not running properly through the joint it is a good plan to gently tap the article which will set up a slight vibration of the joint, and thus assist the spelter to percolate through same. Also, at the same time, the melting spelter should be kept dusted with borax powder.

After brazing, the surplus spelter is removed, the spent borax and oxide scaled off, and in some cases it may be necessary to hammer the point. Excessive hammering should be avoided as the metal at the joint becomes hard and brittle, as at the best it is never as ductile as the rest of the plate.

Where there is danger of the joint cracking under pressure it should always be annealed, so that the metal will be soft and thus stretch somewhat before coming to the breaking point.

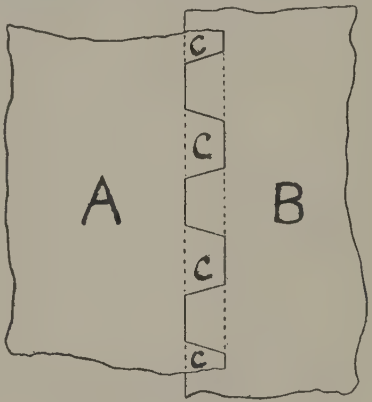

Fig. 4. Brazed Joint in Sheet

Whilst for general purposes it is economical and convenient to use borax as a flux and mix same with spelter, in some cases it is an advantage to use some of the specially prepared paste or fluxed wire mesh which can now readily be obtained.

In some forms of bent joints it is occasionally an advantage to use a strip of brass to fix along the joint so as to ensure that this will melt and flow in the right position.

In obtaining brazing spelters it is always a good plan to state whether they are for iron, copper, or zinc, as the composition varies with the kind of work that is required to be done.

Lesson II will deal with the various methods of welding that are in operation.

MATHEMATICS FOR ENGINEERS

By W. G. Bickley, M.Sc.
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LESSON XXI

SMALL ANGLES

There are certain facts about the trigonometrical ratios of small angles which are of great importance. and which also show the naturalness and usefulness of the radian measure. They are in the nature of approximation formulae, and can be obtained quite easily from Fig. 90. The arc AP subtends an angle θ (radians) at the centre, O, of a circle, and PN and AT are perpendicular to OA. From this it will be seen that if θ is *small*, the lengths of PN, AP, and AT will not be very different, or, if r is the radius, $r \sin \theta$, $r\theta$ and $r \tan \theta$ are nearly equal. That is, for *small* angles, both sine and tangent are very nearly equal to the radian measure of the angle. From tables, we again see that this is true—

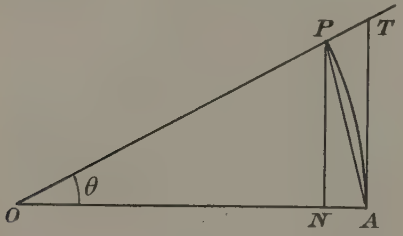

Fig. 90

θ (degrees)	θ (radians)	$\sin \theta$	$\tan \theta$
1	0·01745	0·01745	0·01746
5	0·0873	0·0872	0·0875
10	0·1745	0·1736	0·1763
30	0·5236	0·5000	0·5774

We see here that below 10° the agreement is good, but at 30° the error is 4 per cent for the sine, and 8 per cent for the tangent. We also notice that the sine is less than the radian measure, while the tangent is greater than the radian measure. This means that, in Fig. 90, $PN < PA$, and $AT > AP$; facts which are nearly obvious—the first more so than the second. By considering areas instead of lengths, however, we can make ourselves certain of the order of magnitude of these three, independently of tables. For it *is* evident that

$$\triangle\, OPA < \text{sector } OPA < \triangle\, OPT$$

or $\quad \frac{1}{2}r^2 \sin \theta < \frac{1}{2}r^2 \theta < \frac{1}{2}r^2 \tan \theta$

i.e. $\quad \sin \theta < \theta < \tan \theta$

Divide through by $\sin \theta$, remembering that $\dfrac{\tan \theta}{\sin \theta} = \sec \theta$, and we have

$$1 < \frac{\theta}{\sin \theta} < \sec \theta$$

Now for *small* angles, $\sec \theta$ is only a very little over 1, so we see that $\dfrac{\theta}{\sin \theta}$ cannot differ from 1 by very much when θ is small. Also, the smaller θ is, the nearer to 1 is $\sec \theta$, so that as the angle θ gets smaller, the ratio $\dfrac{\theta}{\sin \theta}$ becomes more nearly equal to 1; symbolically,

$$\text{as } \theta \to 0 \quad \frac{\theta}{\sin \theta} \to 1$$

In the same way, dividing through by $\tan \theta$, we can show that the ratio $\dfrac{\theta}{\tan \theta}$ always lies between 1 and $\cos \theta$, so that

$$\text{as } \theta \to 0 \quad \frac{\theta}{\tan \theta} \to 1.$$

These results prove that, if the angle is small enough, we may use its radian measure for its sine or its tangent, while the table above gives us an idea of the accuracy of this approximation, as the angle gets larger. For use, we may state the formulae

$$\sin \theta \doteqdot \theta, \textit{ and } \tan \theta \doteqdot \theta, \textit{ if } \theta \textit{ is small.}$$

(The sign $\doteqdot$ means "is approximately equal to.")

Examples. 1. What is the angle subtended by a telegraph pole 25 ft. high, at a distance of 150 yd.?

Here, accurately,

$$\tan\theta = \frac{\text{height}}{\text{horizontal distance}} = \frac{25 \text{ ft.}}{450 \text{ ft.}}$$

so that, if the angle is in radians,

$$\theta \doteqdot \frac{25}{450} = \frac{1}{18}$$

In degrees, therefore,

$$\theta \doteqdot \frac{1}{18} \times \frac{180}{\pi} = \underline{3{\cdot}18^\circ}$$

2. Show that, when θ is small, $\sin(\alpha + \theta) \doteqdot \sin\alpha + \theta\cos\alpha$; and obtain an approximation to $\sin 31^\circ$ from the known values of $\sin 30^\circ$ and $\cos 30^\circ$.

We have,

$$\sin(\alpha + \theta) = \sin\alpha\cos\theta + \cos\alpha\sin\theta$$

Now as θ is small, we have $\cos\theta = 1$ and $\sin\theta = \theta$, very nearly, so, using these,

$$\sin(\alpha + \theta) \doteqdot \sin\alpha + \theta\cos\alpha.$$

If $\alpha = 30^\circ$, $\sin\alpha = 0{\cdot}5$, and $\cos\alpha \doteqdot 0{\cdot}8660$; also, $\theta = 1^\circ = 0{\cdot}0175$ radians, so that

$$\sin 31^\circ \doteqdot 0{\cdot}5 + 0{\cdot}8660 \times 0{\cdot}0175$$
$$+\ 0{\cdot}5 + 0{\cdot}0151 = \underline{0{\cdot}5151.}$$

(Tables give 0·5150.)

The cosine of a small angle is very nearly 1, but its difference from 1 is sometimes needed. We are able to get an approximation to the cosine for small angles by means of the foregoing. For $\cos\theta = 1 - 2\sin^2\frac{1}{2}\theta$, and when θ is small, $\sin\frac{1}{2}\theta \doteqdot \frac{1}{2}\theta$, so $\cos\theta \doteqdot 1 - 2 \times (\frac{1}{2}\theta)^2 = 1 - \frac{1}{2}\theta^2$. Thus, even when $\theta = 5^\circ = 0{\cdot}0873$ radians, this formula gives

$$\cos 5^\circ \doteqdot 1 - \tfrac{1}{2} \times (0{\cdot}0873)^2 = 1 - 0{\cdot}0038$$
$$= \underline{0{\cdot}9962}$$

a result agreeing exactly with the four figure tables. In fact, for an angle as big as 30°, the error in this formula is less than ½ per cent.

EXERCISE No. 34

1. A pendulum 30 in. long is oscillating 3 in. on each side of the vertical. Find, approximately, the angle through which it is swinging.

2. Prove that, approximately, $\cos(\alpha + \theta) \doteqdot \cos\alpha - \theta\sin\alpha$, when θ is small, and hence obtain an approximation to $\cos 31^\circ$.

3. Show that the formula $\sin\theta \doteqdot \theta - \frac{\theta^3}{6}$ gives good results for angles as great as 30°. (Later, we may see that we can continue this into a series true for all angles.)

RATES OF CHANGE

The engineer has much to do with rates—one need only mention speed (rate of moving), acceleration (rate of change of speed), horsepower (rate of doing work), revolutions per minute (rate of turning), to make this obvious. When these rates are constant, nothing more than simple arithmetic is needed to deal with them; but in practice they do *not*, in many important cases, remain constant—in such a case as the speed of a piston of a reciprocating engine or pump, for example. To deal with these variable rates, mathematical methods forming the basis of the differential and integral calculus were invented—invented, like so much of mathematics, to enable pressing practical problems to be solved. The remainder of these articles will be devoted to the calculus, as far as space permits, and with a view to those portions of the subject most commonly needed in engineering applications. In this lesson we shall examine carefully the fundamental idea of rates as clear ideas on this are necessary.

The owner of a smart motor-cycle combination was overheard to remark to the passenger in the side-car, as they were bowling merrily along an open stretch of road, "We're doing 60 now. Pretty good, isn't it?" What, exactly, did he mean? 60 miles per hour, of course; but he had to slow down for cross roads within the next minute. So he *did not* mean that he went 60 miles in one hour; and to say that he would have done if he had not slowed down is begging the question, for try to explain speed by words which imply enough knowledge of speed to appreciate what is meant by "not slowing down" does *not* explain it at all. Actually, of course, he had glanced at his speedometer; but this again, is not an *explanation*, any more than one can *explain* what time *means* by looking at one's watch. No, we have to go back to something more fundamental to find an explanation. If you were asked to define speed, it is probable that you would say that speed is "distance gone divided by time taken," or words to that effect. That is perfectly true, but it takes no account of the fluctuations of speed during the time interval. Performing such a division gives us the *average* speed in the interval only. To get a sound idea of *instantaneous speed* we have to follow up this idea, and shall arrive at something slightly different; a speedometer does *not* spend its time in doing long division sums.

From the idea of speed as distance divided by time, how are we to arrive at the correct idea of instantaneous speed? By reducing the interval of time, so that the speed has less chance to alter. Suppose we timed the motor-cycle over a furlong (as in the days of police traps), we should get the average speed in that furlong by division. But the speed might have varied considerably; from a standing start, the average speed would be nothing like the final. But now, suppose we time it over 10 yd., or 1 yd., the speed would have had little chance to change much. (Admittedly, it is not *easy* to actually time a motor-cycle over these small distances with accuracy, but by means of elaborate electrical apparatus, the time taken for a rifle bullet to travel a few yards has been measured sufficiently accurately for its speed to be found to within a few feet per second; there is no *logical* difficulty in supposing such a measurement possible.) For *practical* purposes these last determinations would be all that is required, but in fact, strictly speaking, we have still found nothing more than an *average* speed; but over so small an interval that the speed has varied so little that the variation is insignificant from the practical standpoint. Such an approximation is not enough for the more exact requirements of mathematics, and, with a formula, there is no difficulty beyond the labour of calculation to prevent our dealing with much smaller time and distance intervals. We then calculate the average velocity for progressively smaller intervals, and find what the velocity *tends to* as the intervals *tend to* zero. (Note here, once and for all, that it is *nonsense* to say that the intervals "become equal to" zero. In no time, the body goes no distance, no matter *how* fast it moves.)

We will try to make the process clear by numerical work for a falling body. In this case, if s represents the distance fallen, in feet, in t seconds, we have, sufficiently accurately for our purpose,

$$s = 16t^2.$$

This equation gives the whole history of the fall—where the body is at *any* time—and so will give us sufficient information to find the velocity at any time, say, at the end of 2 sec. At the end of 2 sec. ($t = 2$) the body has fallen $16 \times 2^2 = 64$ ft. To find speed we have to follow its motion a little farther, that is, to find the distance moved in a small interval, and then decrease this interval progressively, keeping our eye on the speeds to see to what they are tending. Thus, taking an interval of 0·1 sec., and putting, therefore, $t = 2{\cdot}1$, we find the total distance fallen in 2·1 sec. is $16 \times 2{\cdot}1^2 = 70{\cdot}56$ ft. We now find the distance travelled in the tenth of a second by subtracting the 64, getting 6·56 ft. The average velocity in this tenth of a second is (distance/time) $\frac{6{\cdot}56}{0{\cdot}1} =$ 65·6 ft./sec. Now to take a smaller interval, say, 0·01 sec. When $t = 2{\cdot}01$, $s = 64{\cdot}6416$ ft. Subtracting the 64, we have as the distance travelled in the hundredth of a second, 0·6416 ft. So the average velocity in the interval is $\frac{0{\cdot}6416}{0{\cdot}01} = 64{\cdot}16$ ft./sec. Proceeding in this way, and carrying out all the calculations with absolute accuracy, we get results as tabulated—

Interval	s	Distance in Interval	Average Speed
0·1	70·56	6·56	65·6
0·01	64·6416	0·6416	64·16
0·001	64·064016	0·064016	64·016
0·0001	64·00640016	0·00640016	64·0016

We have probably gone far enough to make it appear at least probable that, by further decreasing the interval, we shall approach nearer and nearer to 64, and it is this 64 which is the *instantaneous* speed when $t = 2$. Notice that, however small we make the interval, the speed is never *equal to* 64; 64 is only obtained as a *limit* to which the calculated speeds tend.

This limiting process is fundamental in the mathematical study of rates of change, whether of speeds or of any other rate, and the student should not pass on until he understands and appreciates the points raised.

It will be realized from the above that in these questions involving rates of change, we shall be dealing very frequently with small intervals, or with small increases, or with small differences, and so a symbol to denote these will be useful. The one usually adopted is the Greek letter δ (the Greek d, called "delta," d being the initial letter of the word "difference"). Thus δt means "a small interval of time," and δs, "a small distance." (Note that δ is *not* a *quantity*—we cannot multiply or divide by δ, nor can it be cancelled.) δt has to be taken together as meaning "a small interval of time," and similarly with δs. Then, for average velocity, we must have $\frac{\delta s}{\delta t}$. It is then convenient to have a separate symbol to denote instantaneous velocity (or instantaneous rate of change in general), and in this the Greek δ is replaced by the English d. Thus, the symbol for instantaneous speed in the above (i.e. instantaneous rate of change of s with respect to t) is $\frac{ds}{dt}$.

In the next lesson we shall go farther into the way of finding $\frac{\delta s}{\delta t}$ and $\frac{ds}{dt}$, and similar rates of change.

ANSWERS TO EXERCISES 32 AND 33

(32)

1. $\frac{1}{[illegible]}$.
5. (a) 88°38′, 54°36′, 36°46′, (b) 42°48′, 32° 12′, 28·29.

(33)

1. 11·52
2. 15·71 in., 94·25 sq. in.
3. 0·1612 sq. in.
4. 7·25 sq. in.
5. 182·4 cub. yd.

PIONEERS OF ENGINEERING

By J. F. Corrigan, M.Sc., A.I.C.

LESSON XIX

THERMO-DYNAMICS FOUNDED

SADI CARNOT (1796–1832)

To Sadi Nicolas Léonhard Carnot, the son of Napoleon's famous General of that name, is due in a very large measure the credit of being the first investigator to enunciate clearly the important law of thermo-dynamics respecting the mechanical power of heat.

Carnot compared the mechanical power of heat to the power possessed by a head of water, and, as in the latter case, mechanical work may be done by the water's fall from a position of high level to one of lower level, so also may work be performed when a substance is allowed to fall from a higher to a lower temperature. Carnot showed that this substance whose fall in temperature could give rise to mechanical work might be a gas, a liquid, or a solid, and he pointed out also that the amount of work obtained from the heat via the "working" substance is independent of the nature of the substance.

In short, Carnot clearly conceived the idea of a cycle of processes or operations through which a substance could be passed. He pointed out that when a body is subjected to changes of pressure, temperature, and volume, and, after undergoing this series of changes, is ultimately brought back to its original state of pressure, temperature, and volume, the contained energy of that body is equal to its original energy before it was put through the series or cycle of operations.

Sadi Carnot

Here, therefore, arose the basic conception of a cycle of changes through which a substance could be placed, a conception which, of course, is of the utmost importance in all heat and internal combustion engine theory. The particular cycle of operations which bears Carnot's name refers to the cycle of changes through which an ideal substance, such as a perfect gas, may be put. Other investigators have worked out more practical types of cycles of operation which can be applied individually to steam, hot air, and internal combustion engines.

Carnot, whose first conception of a series of cyclic operations has proved to be of so much theoretical importance in the engineering world, was born at Paris in 1796. Educated at the École Polytechnique of that city, he left that Institution in the year 1812, and two years later obtained a commission in a French Corps of Engineers. Owing, however, to political circumstances connected with the downfall of his father, L. N. M. Carnot, he resigned from the Corps. In 1818 he joined up again with the military forces, his appointment being made to the Staff Corps (*état-major*) in which body he was given a lieutenancy.

His health failed him, however, and he again resigned from the army service, this time to give himself up entirely to the prosecution of scientific and philosophical research.

Carnot was really a most brilliant individual. Skilled in many branches of science, including physics, chemistry, natural history, and even political economy, he was an artist as well, combining with his scientific activities, the practice of music and painting. Sport occupied a portion of his time also, and, as an athlete, he gained a considerable reputation in the pastimes of swimming, fencing, and archery.

In abstruse philosophical matters, Carnot proved to be one of the most original and profound thinkers of his day. But his numerous activities brought him to an early grave. Like an ill-regulated flame, his life burned rapidly to a close. After much ill-health during the later portion of his life, he died at Paris on 24th August, 1832, from an attack of brain fever followed by cholera.

OUTLINES OF AUTOMOBILE ENGINEERING

By J. L. Milligan, B.Sc. (Vict.), M.I.A.E.

LESSON V

ENGINE CONSTRUCTION—(*contd.*)

Camshafts. Camshafts have to revolve once for each cycle of operations of the engine, i.e. two revolutions of the crankshaft to one of the camshaft. Both chain drive and gear drive are in use for this purpose; chain is the quieter, and is therefore desirable, but it occupies more length than gears. It is more limited in its application, as the centres of the wheels are fixed by the number of teeth in the chain wheels and length of chain. The size of wheels may be determined by some feature of engine design, and the chain may not design in nicely. On the other hand, if the camshaft is high up in the engine, the larger of the two gears becomes a difficulty, and an intermediate wheel has to be used to obtain reasonable size with low tooth velocity. This introduces another trouble, as there are two wheels whose mesh can vary. Too much fitting of crank bearings may raise the shaft and destroy the accuracy of centre distances between gears. Sometimes the chain is passed round three gears, as shown in Fig. 22, the third wheel driving a magneto or dynamo which can be slid slideways to adjust the chain. Chains are always of the inverted tooth or noiseless type. Gears are sometimes made of non-metallic material to avoid noise. Paper, linen, or vegetable fibres are impregnated with binding material and subjected to pressure and, in some cases, heat. The resultant material is then machined as if of metal.

There is no question that the valve gear drive should be at the back end of the engine as close to the fly-wheel as possible, so that the irregular drive of the cams should not have to pass through the crankshaft, but is steadied by the fly-wheel. The only reason for putting gears in front is to make them accessible.

The drive for overhead camshafts sometimes takes the form of a vertical shaft driven by spiral gears on the crankshaft, driving the overhead camshaft or two overhead camshafts by spiral or bevel gears. Such gears are seldom quiet as they are usually at the front end, and subject to the crank vibrations. A very long chain with a special method of taking up slack is in successful use on several makes of overhead valve engine.

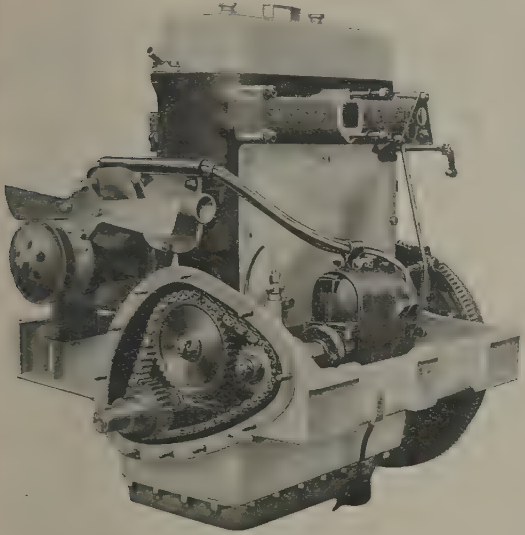

FIG. 22

THE SLEEVE VALVE

There are two types of sleeve valve in use, viz., the single-sleeve and double-sleeve.

If in a cylinder, such as Fig. 23, we have a loose head like a fixed piston and reciprocate the cylinder at half the engine speed over a travel double the height of ports *a*, the ports or openings *a* would be opened and closed each during a whole crank revolution. If both the main crank and cylinder operating crank are at the top together, the cylinder ports would remain open during one down and one up stroke of the piston. If a fixed outer casing containing induction and exhaust pipes surround this cylinder, both inlet and exhaust will be open to the cylinder together. So far we have considered one moving cylinder (called a sleeve, the outer casing being called the cylinder, though not truly the working cylinder). The "single-sleeve" engine times the opening of inlet and exhaust ports by partially rotating the sleeve during two main crank revolutions, so that the sleeve ports are opposite the exhaust port while the main piston is rising and sleeve descending, and opposite the inlet port during the following stroke of the piston. A point on the sleeve travels in an almost circular path.

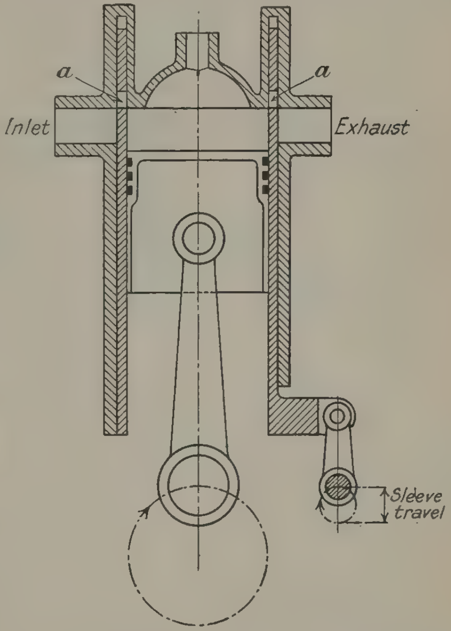

FIG. 23. DIAGRAMMATIC VIEW OF SLEEVE ENGINE

In the case of the double-sleeve engine (Fig. 24), there is a second sleeve between the inner sleeve and the main block carrying the piping, etc., and this is operated by a second crank nearly at right angles to the inner sleeve crank and following it. When the inner sleeve is going down the lower half of its travel and the ports are open below the head, the outer sleeve is coming down the upper part of its travel and a port on the exhaust side is open to the exhaust port on the inner sleeve and to the cylinder port. In the lower half of the following up stroke of the sleeve and down stroke of the piston, the ports are still open in the inner sleeve, but the outer sleeve is still travelling down and a port on the inlet side is open to the inlet pipe.

The sleeves in the single-sleeve engine are operated by a shaft running at half engine speed, which drives by special gears into short cross-shafts, one to each valve, or by an "angled crank" shaft. In the double-sleeve engine there is a valve shaft with two small cranks to each cylinder.

There is still considerable controversy with regard to the merits of sleeve engines. They have set a high standard of silence and of endurance, due to the absence of poppet valves and springs. Owing to the shape of the combustion chamber much higher compressions can be carried without pinking than with side valves, and there is an absence of carbonizing with its attendant cost of cleaning. One drawback is the somewhat higher first cost and higher oil consumption. During recent years, both in Europe and America, great changes have occurred in the manufacture of both types of sleeve engines, and the competition is likely to be more severe than heretofore.

Owing to the smoother drive, chains on sleeve valve engines are not subject to the same shocks as on poppet valve engines, and are almost always without adjustment.

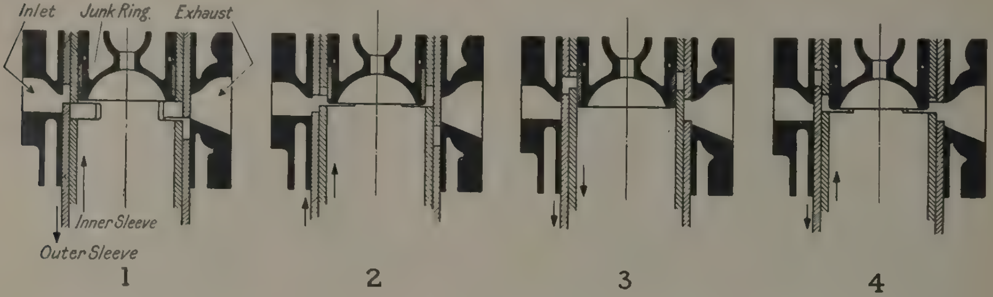

FIG. 24

Diagram 1 shows the inlet port opening and the exhaust port closing at the beginning of the suction stroke. The inner sleeve is rising and the outer sleeve is descending.

Diagram 2 shows the exhaust closed and the inlet closing at the beginning of the compression stroke. The ports in the inner sleeve are passing up into the cylinder head, where they are sealed by the junk ring during the period of maximum pressure and temperature at the beginning of the firing stroke (see Diagram 3).

Diagram 4 shows the exhaust already well opened at the beginning of the exhaust stroke. The sleeves are now again moving to the position shown in Diagram 1.

The complete cycle of operations—suction, compression, firing, and exhaust—occupies two revolutions of the crankshaft.

FUELS

Petrol. The commonest fuel is an indeterminate mixture of hydrocarbons, known as motor spirit or petrol.[1] It is obtained by the distillation of crude petroleum or mineral oil. There are many brands which vary in composition. It usually has a density of ·740 to ·760 times that of water, and the lighter constituents evaporate at comparatively low temperature, so that its density increases slightly if much exposed to the air in warm weather.

It is a Highly Dangerous Liquid to Handle. There are many legal requirements regarding it which are relaxed when it is used in the tank of a vehicle. Familiarity has too often bred contempt of its dangers, and many lives have been lost through carelessness in handling it. No naked light should be brought near it.

Benzole, which is a by-product of the carbonizing of coal, either in a coke oven or at a gas works, is also used, seldom alone but generally mixed with petrol. Benzole will freeze and is very difficult to vaporize at low temperatures. On the other hand, it does not pink like petrol until higher compressions are used. It is more economical in use, and is now sold at prices comparable with petrol.

Ordinary Paraffin Lamp Oil has been used, but requires much care in its vaporization and, unless the cost of spirit is very greatly in excess of that of paraffin, the latter stands little chance of use, even on commercial vehicles.

Alcohol, either mixed with petrol or with benzole, is extensively used on the Continent as fuel. The great extent to which the revenue of Great Britain is derived from taxes on potable alcohol has prevented the development in this country, owing to the fear that such spirit would be used in a dutiable manner.

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[1] *Petrol* is the trade-mark of one manufacturer, but it has come into colloquial use for all. Other makers use the term Motor Spirit.

During the war, coal gas, both compressed and uncompressed, was tried, and in case of any greatly increased cost of liquid fuel may re-enter the field as a competitor.

CARBURETTORS

The simplest type of carburettor is as shown in Fig. 25. Air passes to the engine through a smooth reduced pipe called a Venturi tube. Fluids in motion possess energy due to their velocity (kinetic energy) and to their pressure (potential energy). If by reducing the bore of the pipe (as in the Venturi tube) the velocity increases, the total energy remains the same, but the kinetic energy having been increased, the potential energy is decreased and the pressure is reduced; this is a well-known theorem in hydraulics.

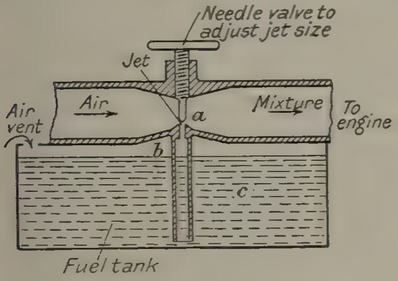


FIG. 25

As the pressure at *a* is less than at *b*, which is atmospheric pressure, the fuel in *c* will be blown by the atmospheric pressure above it into the Venturi tube and mixed with the air. The varying speeds of the engine cause variation in the speed of the air and the petrol flow.

If such arrangement as Fig. 25 is used (as it is on some cheap agricultural engines), it is found that the proportion of petrol to air rises as the air velocity rises, so that a jet which supplied enough petrol at 1,000 rev. per min. will supply too much at 2,000 r.p.m. Also at starting, the air speed when turning the engine slowly will not lift any petrol. To control the engine speed, throttle valves are now in universal use. They are simple shutters in the induction pipe, or rotating valves, which regulate the supply of mixture, and are usually combined with the carburettor as they are often used to control the quality as well as quantity of the mixture.

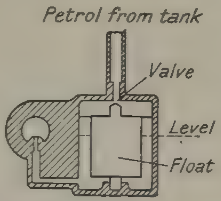


FIG. 26. SIMPLE FLOAT VALVE

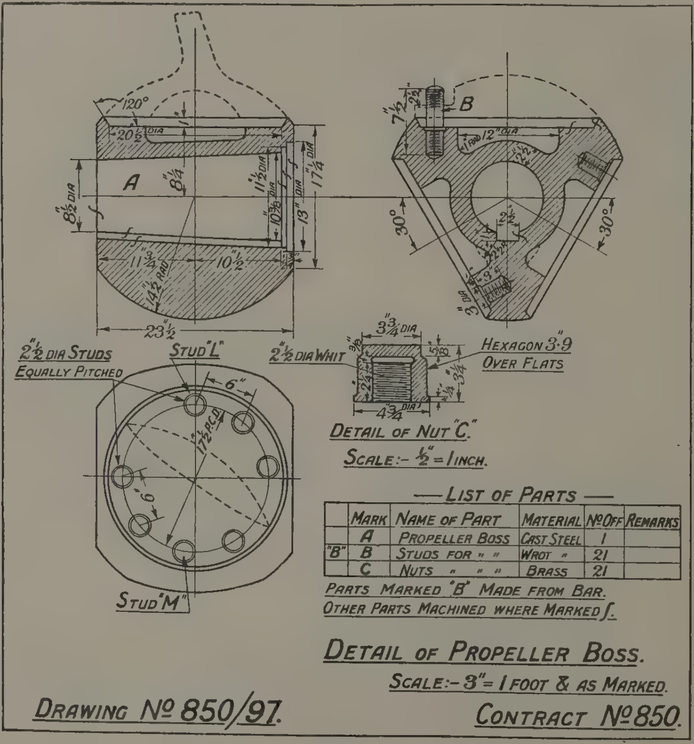
In order to maintain a constant level in the chamber feeding the jet, it is made as shown diagrammatically in Fig. 26, a float and needle valve closing the inlet when the level is at a determined position, usually just below the jet orifice.

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XX

Checking. We have already suggested that the test of a good machine drawing is to send it to the shops ordering the details embodied on the drawing. If the drawing is good the craftsman will make the parts as required without query or help from the drawing office. If the drawing is poor then we may expect requests for further information, or possibly a finished part failing to take its place in the machine. It follows that, to meet the above requirements of a good drawing it is necessary, when making the drawing, to look at the detail from more than one angle. Moreover, of the various viewpoints, the practice of the workshops is not the least important.



	MARK	NAME OF PART	MATERIAL	Nº OFF	REMARKS
	A	PROPELLER BOSS	CAST STEEL	1	
"B"	B	STUDS FOR " "	WROT "	21	
	C	NUTS " " "	BRASS	21	

PARTS MARKED "B" MADE FROM BAR.
OTHER PARTS MACHINED WHERE MARKED ʃ.
DETAIL OF PROPELLER BOSS.
SCALE:- 3" = 1 FOOT & AS MARKED.
DRAWING Nº 850/97.
CONTRACT Nº 850.

Fig. 1

It is evident, from the above, that a certain amount of self-criticism is necessary when making a machine drawing. Further, in criticizing a drawing we ought to keep within practical limits. Remove the limitations of shop practice, and any drawing, however good, is not beyond criticism. If we remember that the true function of criticism is creative, and not destructive, we may end our study in acquiring the technique of good machine construction and drawing.

Interference. One of the most common and often the least obvious of errors made in machine drawing is the interference of machine parts. Fig. 1 shows a cast-steel boss for a three-bladed propeller. It will be seen that each blade is secured to the boss by seven studs. Further, it will be observed that the studs for each blade are arranged in the same position relative to the centre line of the blade. It follows that the three studs marked *L* lie in the same vertical plane, and the three studs marked *M* occupy a common vertical plane. The sectional end-elevation shows the studs marked *L*, but does not show the studs *M*. We are, therefore, without a view showing the proximity of the studs *L* and *M*. If we draw an outside end-elevation of the boss showing the above studs to dotted lines, Fig. 2, then we observe an apparent interference of the studs. The above apparent interference would exist if the studs were arranged in the same vertical plane. To check for interference we may draw a large scale outline of the two adjacent studs, as in Fig. 3. Proceeding with an end view giving an outline of the plane *ab*, taken through the stud *L* together with an end view of the adjacent stud *M*, we find an interference of view to the extent of the closely shaded area. This is obviously an error, and to avoid the above interference it will be necessary to re-arrange the studs. A suggested correction is shown in Fig. 4.

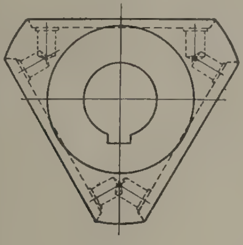


FIG. 2

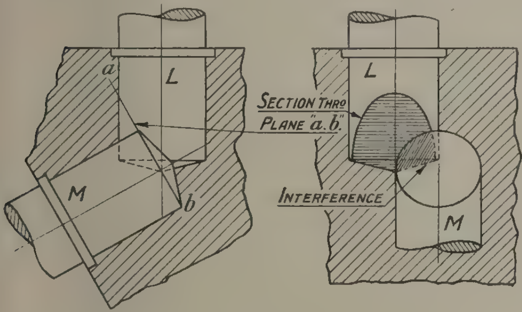


FIG. 3

Correspondence. A drawing may be quite correct within itself and yet leave the detail failing to agree with the adjoining machine part.

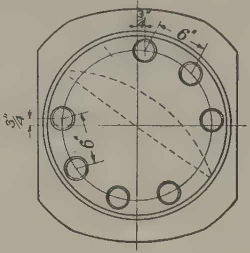


FIG. 4

It follows that a drawing must often be considered as component to a second drawing and, as such, requires a mutual criticism. An example of the above point is given in Fig. 1, where the taper of the cone in the boss is given in terms of a difference in diameter for the length of boss. The adjoining part in this case is the propeller shaft (Fig. 5) and, the lengths of cone being different, this form of dimensioning would make it difficult to obtain the degree of accuracy required for this important fit. The more convenient form of dimensioning is to give the angle of the cone in terms of a taper, e.g. taper = one in twelve on diameter. Adopting this method for both the boss and the adjoining shaft only one diameter need be given in each case, and the difference in length of cone in the two adjoining parts does not now necessitate a careful determination of the second diameter.

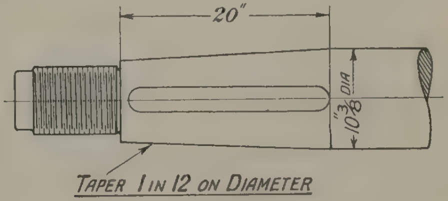
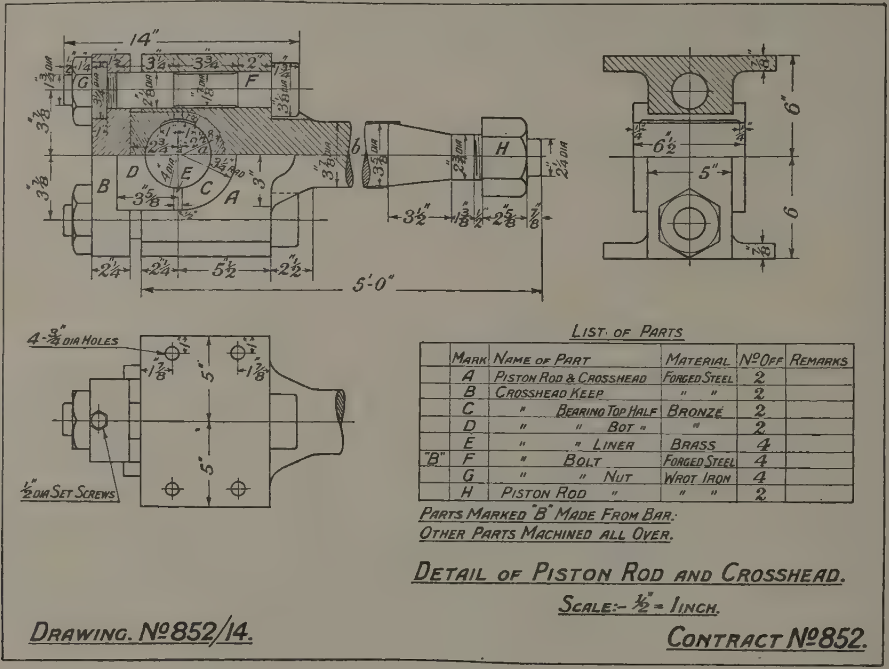


FIG. 5

Omission. As a further example in checking we will take the detail drawing of the piston-rod and crosshead shown in Fig. 6. We may first examine the drawing for completeness. The omission of information may prove both time wasting and annoying to the shops, and freedom from this fault is most desirable. We must distinguish between the common and the exceptional in deciding what may be omitted from a drawing. In the above drawing it will be observed that no detailed dimensions are given either for the crosshead bolt nuts or the usual provision, in the form of a split pin, against the slacking back of the nut. The former omission may be considered to be in order if the nut is a standard design of which the shops have particulars. Regarding the latter omission it is not probable that the size of split pin will be standardized, and so this omission is not allowable. An omission of a technical nature occurs in the lack of provision for lifting the piston-rod, a requirement when overhauling the engine. To correct this omission we may show a hole in the rod end tapped $\frac{7}{8}$ in. diameter, $1\frac{1}{4}$ in. deep, to take an eyebolt. A further technical omission occurs in the absence of provision for the supply of oil to the bearing. This omission may be rectified by adding the sectional view of the rod, as shown in Fig. 7.



LIST OF PARTS

	MARK	NAME OF PART	MATERIAL	Nº OFF	REMARKS
	A	PISTON ROD & CROSSHEAD	FORGED STEEL	2	
	B	CROSSHEAD KEEP	" "	2	
	C	" BEARING TOP HALF	BRONZE	2	
	D	" " BOT "	"	2	
	E	" " LINER	BRASS	4	
"B"	F	" BOLT	FORGED STEEL	4	
	G	" " NUT	WROT IRON	4	
	H	PISTON ROD "	" "	2	

PARTS MARKED "B" MADE FROM BAR:
OTHER PARTS MACHINED ALL OVER.

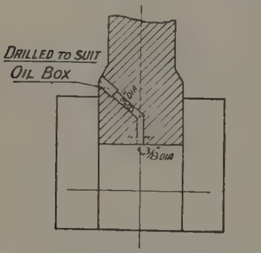
DETAIL OF PISTON ROD AND CROSSHEAD.

SCALE:- ½" = 1 INCH.

DRAWING. Nº852/14.

CONTRACT Nº852.

FIG. 6



SECTION THRO "a.b." (FIG.6)

FIG. 7

We also observe that the dimension of the shank where it enters the crosshead body suggests a rectangular instead of a circular section. For completeness, the corresponding diameter should be substituted for this dimension.

In addition to the above we observe an error common to the previous example, Fig. 1, i.e. the taper of the piston-rod is given in terms of two diameters over a length of rod. Again, the safer and more convenient method of dimensioning would be to specify a taper of one in four on diameter.

EXERCISE

We advise the student to make working drawings of the two details, Figs. 1 and 6, corrected in accordance with the above criticism.

APPLIED MECHANICS

By Geo. W. Bird, Wh.Ex., B.Sc., A.M.I.Mech.E., A.M.I.E.E.

LESSON XX

ELEMENTS OF MECHANISM

Work. We often speak of having performed work when we are physically or mentally tired, but the engineer's use of the term work is of more limited application. Work is only done when a force, acting upon a body, produces motion of the body. When the body moves in the direction in which the force acts, the amount of work done is given by the product of the force acting and the movement of the body. If, however, the movement of the body is not in the direction in which the force acts, Fig. 110, then the work done is measured by the product—

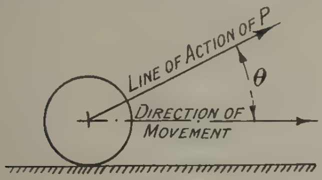


Fig. 110

Force acting × movement of body × cosine θ . . . . (1)

where θ is the angle between the line of action of the force and the direction in which the body moves.

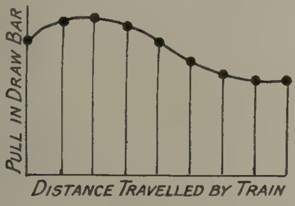


Fig. 111

If the force is a variable one, the work done by its application is conveniently found from a diagram, in which are plotted the forces acting and the corresponding distances which the body has moved from some agreed starting-point. As an example, we may consider the work done by a locomotive in drawing a train. The drawbar pull may vary considerably, depending upon the weight of the train, the various frictional resistances, the gradient, etc., but if we know the drawbar pulls for a series of train positions, we can plot these, as in Fig. 111, and so obtain a diagram, the area of which represents the work done by the locomotive between the two extreme train positions considered.

Example. A trial of a locomotive for the purpose of recording the tractive efforts at a series of distances from the start yielded the following results—

Tractive effort lb. .	3,710	3,100	2,500	1,795	1,480	1,100
Distance from start yd.	—	180	205	280	360	540
Tractive effort lb. .	900	300	235	205	150	—
Distance from start yd.	800	1,200	1,600	1,800	2,240	—

These values of tractive efforts and distances are plotted in Fig. 112, and give us a diagram *ABCD*, the area of which represents the total work done by the locomotive in hauling the train a distance of 2,240 yd. The area *ABCD* can be obtained by means of the planimeter, and using the scales employed for distance and effort respectively we find that the total work done by the locomotive is 6,075,000 ft.-lb.

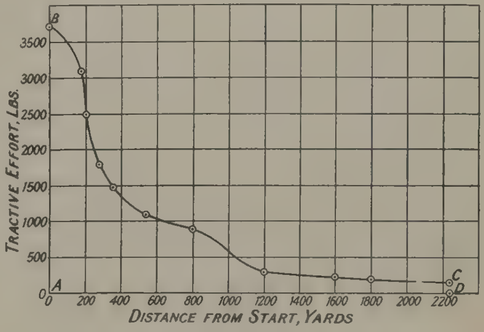


Fig. 112

Circular Motion

If *A* (Fig. 113) represents a body which is constrained to move in a circular path when acted upon by a tangential force P, then the work done by the force P in turning the body through the angular displacement θ

$$= P \times \text{arc } A_1A_2 = P \times R \times \theta$$
$$= \text{torque} \times \text{angular displacement of body.}$$

If the torque is constant, the calculation of the work done is simple, but if, as we usually find in practice, the torque is varying, then the total work done can be estimated by plotting a series of values of torques and the corresponding angles turned through, the net area of the figure so obtained represents the amount of work done. An instance of this kind occurs in the reciprocating engine, in which the turning moments or torques applied to the crankshaft when plotted to a crank angle base give a diagram of the type shown in Fig. 114.

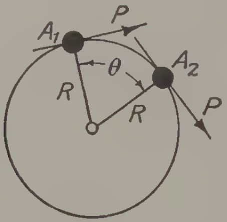


FIG. 113

POWER

In estimating the capacity of a prime mover or machine for a particular purpose, we find it necessary to take into account the element of time; we wish to know, not only the *total* amount of work done, but the *time interval* during which this work is done. This introduces the notion of a *time rate of doing work*, which we term *Power*. The usual *mechanical unit of power* is 1 h.p., which is equal to 33,000 ft.-lb. of work performed in a time interval of 1 minute. The *electrical unit of power* is a kilowatt, and since 746 watts = 1 h.p.

$$\therefore \text{ 1 kw.} = \frac{1000}{746} = 1{\cdot}34 \text{ h.p.}$$

ENERGY

When work is to be done, we find it necessary to call upon some body which either possesses, or can be given, a store of energy. Fuels of all kinds, water under suitable circumstances, wind, etc., are examples of sources of energy of which the engineer makes frequent use, transforming and transmitting the supply to suit his needs. We shall, at this point, restrict our study of energy to the two forms, *Potential Energy* and *Kinetic Energy*, using these terms in a mechanical sense.

POTENTIAL ENERGY

The store of *potential energy* which a body possesses comes about in virtue of its position relative to some other body, or, as in the case of a strained spring, the *potential energy* store comes about by one part of the spring being forced into a position nearer to or farther away from the other part of the spring against a variable force produced by internal stresses in the material of the spring. Alterations in lengths and shapes of bodies due to applied forces represent, under certain conditions, stores of energy, but we seldom use these energy stores for the purpose of doing work, referring to them more often as cases of *strain energy*. A reservoir of water situated in some elevated position amongst the hills, or a tank of water on the top of a tall building furnish us with an example of a body, water in this case, which, because of its position, possesses a store of potential energy. By a suitable arrangement of pipe lines we can lead the water down to a turbine and so cause it to do work for us. Neglecting, for the time being, all frictional and other losses, we may say that every pound of water, as it arrives at the turbine brings with it a store of energy = 1 lb. × h ft. (lb. and ft. being our usual units), and this will be due to the original position of the water having been h ft. above the level of the turbine. Our expression for potential energy then is

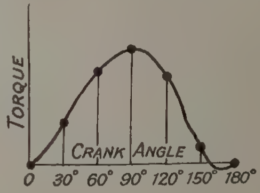


FIG. 114

$$\text{potential energy} = W \times h$$

and is conveniently measured in foot-pounds.

EXAMPLE. The monkey of a pile driver weighs 0·25 ton, and, when in action, is allowed to fall vertically a distance of 6 ft. before coming into contact with the head of a pile which it drives a depth of 8 in. Determine the average resistance offered by the pile.

Total distance moved by monkey

$$= 6\tfrac{2}{3}\text{ ft.}$$

$\therefore$ loss of potential energy

$$= W \times h$$

$$= 560\text{ lb.} \times 6\tfrac{1}{4}\text{ ft.}$$

and this is available for the purpose of doing work in driving the pile ; hence

$$W \times h = \text{average resistance of pile} \times \text{movement of pile}$$

$$\text{or } 560 \times 6\tfrac{2}{3} = \text{average resistance of pile} \times \frac{8}{12}$$

$\therefore$ average resistance of pile

$$= \frac{560 \times 20 \times 12}{3 \times 8} = 5{,}600\text{ lb.}$$

KINETIC ENERGY

A body in *motion* possesses a store of energy which we term *kinetic*, and in bringing the body to rest we can cause the body to do an amount of work equivalent to its store of kinetic energy. Conversely, in setting the body into motion, it is necessary to give to it the same amount of energy which it will then store as kinetic energy. We have shown in Lesson XVIII, that $v^2 = 2gx$

where v is the velocity of a body which has fallen freely, or, has been subjected to a constant velocity-acceleration of g whilst moving a distance of x ;

$$\therefore \frac{v^2}{2g} = x \quad . \quad . \quad . \quad . \quad (1)$$

Also we know, that, in accordance with the principle of the *conservation of energy*, a body of mass W, in falling a vertical distance x, *reduces* its store of *potential energy* by an amount Wx, and *increases* its store of *kinetic energy* by an *equal* amount ; hence, multiplying each side of (1) by W we may write—

$$\frac{Wv^2}{2g} = Wx$$

Thus, we obtain the expression, $\frac{Wv^2}{2g}$, for the *kinetic energy* store of a body whose weight is W, when moving with a velocity v.

If the motion of the body is linear, then the expression $K.E. = \frac{Wv^2}{2g}$, is in a convenient form, but, in cases of circular motion, fly-wheels, rotors, etc., the expression can be put into a more convenient form as follows—

$$\text{K.E.} = \frac{Wv^2}{2g}$$

$$,, \quad = \tfrac{1}{2} . \frac{W}{g} \times v^2$$

$$,, \quad = \tfrac{1}{2} M \times \omega^2 k^2$$

$$,, \quad = \tfrac{1}{2} I \omega^2$$

K.E. = store of kinetic energy in ft.-lb. (usual units).
v = linear speed of body in feet per sec., measured at a distance " k " from the centre of rotation.
k = radius of gyration of rotating body.
ω = angular speed in radians per second.

TOTAL K.E.

In the case of, say, a fly-wheel of a stationary engine, the centre of rotation is a fixed point and the K.E. store is given by the expression, $\frac{Wv^2}{2g}$; if, however, the centre of rotation of the wheel is itself in motion, as in the case of, say, the wheels of a locomotive, then the wheel contains a further store of K.E. $= \frac{W{v_2}^2}{2g}$; where v_2 is the linear velocity of the wheel centre, and hence the total K.E. $= \frac{Wv^2}{2g} + \frac{W{v_2}^2}{2g}$, which may also be written

$$\text{total K.E.} = \tfrac{1}{2}Iw^2 + \frac{W{v_2}^2}{2g}.$$

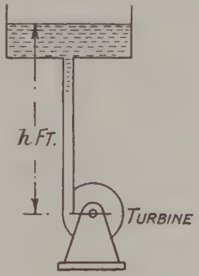


FIG. 115

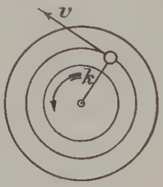


FIG. 116

EXAMPLE. A solid C.I. disc fly-wheel is 4 ft. 6 in. diameter and 6 in. thick. Calculate the K.E. stored in the wheel at 350 r.p.m. 1 cub. in. of C.I. weighs 0·26 lb.

We must first find the radius of gyration k, treating the wheel as a circular lamina, as follows—

$$I = Ak^2$$

$$\therefore k^2 = \frac{I}{A} = \frac{\pi}{32} D^4 \times \frac{4}{\pi D^2} = \frac{D^2}{8}$$

Where A = area of a circle 4 ft. 6 in. diam.
and I = polar moment of inertia of this circle.

Weight of the wheel $= W = \frac{D^2\pi}{4} \times$ thickness $\times$ weight per unit volume $= \underline{3{,}575}$ lb.

$$\text{K.E.} = \tfrac{1}{2} I\omega^2 = \tfrac{1}{2} Mk^2\omega^2$$

$$,, \quad = \tfrac{1}{2} \frac{W}{g} k^2\omega^2$$

$$,, \quad = \tfrac{1}{2} \times \frac{3575}{32\cdot3} \times \frac{4\cdot5 \times 4\cdot5}{8} \times \left(\frac{350 \times 2\pi}{60}\right)^2$$

$$,, \quad = \underline{188{,}800} \text{ ft.-lb. stored in the fly-wheel.}$$

EXAMPLE. The fly-wheel, mounted on the main shaft of a marine oil engine, weighs 5 tons, and may be regarded as the equivalent of a solid C.I. disc of 6 ft. 6 in. diameter. Calculate the total store of K.E. in the wheel when the engine is running at 81·75 r.p.m., and the motor vessel is making 11·68 knots.

The total K.E. will be the sum of the K.E. due to the rotation of the wheel about its own axis together with the K.E. due to the movement of the fly-wheel as a part of the vessel; or,

$$\text{total K.E.} = \tfrac{1}{2} I\omega^2 + \frac{Wv_2^{\,2}}{2g}.$$

As in the previous example, $k^2 = \frac{D^2}{8}$

$$= \frac{42\cdot25}{8} \text{ (ft.)}^2 \text{ units} \quad . \quad . \quad . \quad . \quad (1)$$

$$\omega^2 = \left(\frac{81\cdot75 \times 2\pi}{60}\right)^2 = (2\cdot725\pi)^2 \quad . \quad (2)$$

$$v_2^{\,2} = \left(\frac{11\cdot68 \times 6080}{60 \times 60}\right)^2 \text{(ft./sec.)}^2 \; . \quad . \quad (3)$$

Substituting (1), (2), and (3) in the following expression

total K.E. $= \tfrac{1}{2} I\omega^2 + \frac{wv_2^{\,2}}{2g}$; we obtain

total K.E.

$$= \left[\tfrac{1}{2} \times \frac{5 \times 2240 \times 42\cdot25 \times (2\cdot725\pi)^2}{32\cdot2 \times 8}\right] + \left[\tfrac{1}{2} \times \frac{5 \times 2240}{32\cdot2} \times \left(\frac{11\cdot68 \times 6080}{60 \times 60}\right)^2\right]$$

$$\text{total K.E.} = 67{,}330 + 67{,}690$$

$$,, \quad ,, \quad = \underline{135{,}020} \text{ ft.-lb.}$$

EXAMPLE. An engine develops 350 h.p. when running at 180 r.p.m. If the fluctuation of energy in the fly-wheel is equal to $\frac{1}{10}$ of the work done by the engine per revolution, and the fluctuation of speed is ± 1 per cent of the mean speed, determine the necessary weight of the fly-wheel, the radius of gyration of which is 3·2 ft.

$$\text{Work per revolution} = \frac{350 \times 33{,}000}{180}$$

$$\tfrac{1}{10} \text{ work per revolution} = \frac{350 \times 33{,}000}{10 \times 180}$$

$$= \frac{19{,}250}{3} \text{ ft.-lb.}$$

and this is the difference between the maximum and minimum K.E. stored in the fly-wheel, or, $\text{K.E.}_{max} - \text{K.E.}_{min}$.

$$\text{Mean speed} = \frac{180 \times 2\pi}{60} = 6\pi \text{ radians per sec.}$$

Fluctuation in speed $= \pm \cdot01 \times 6\pi$ radians per sec.

$\therefore$ maximum speed $= 1\cdot01 \times 6\pi$ radians per sec.

and minimum speed $= 0\cdot99 \times 6\pi$ radians per sec.

$$\text{K.E.}_{max} - \text{K.E.}_{min} = \tfrac{1}{2} I[(\text{max. speed})^2 - (\text{min. speed})^2]$$

$$\frac{19{,}250}{3} = \tfrac{1}{2} \times \frac{W}{32\cdot2} \times 3\cdot2 \times 3\cdot2 \; [(1\cdot01 \times 6\pi)^2 - (0\cdot99 \times 6\pi)^2]$$

$\therefore W = 2{,}838$ lb. weight of fly-wheel.

STEAM TURBINES

By W. J. Kearton, M.Eng., A.M.I.Mech.E., A.M.I.N.A.

LESSON VII

CONSTRUCTIONAL DETAILS OF IMPULSE TURBINES

Nozzle Box. This detail is constructed in an almost endless variety of forms, and its arrangement and shape are influenced to some extent by the system of governing adopted.

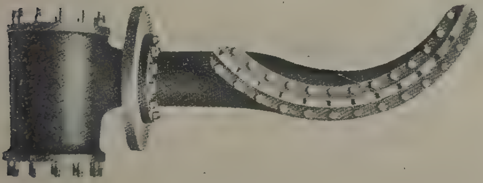


Fig. 25. Cast Steel Pipe Section Nozzle Box

When superheated steam is used it is frequently made a separate steel casting, as shown in Fig. 25, which shows a nozzle box or steam chest for a Metropolitan-Vickers turbine. The nozzle box is only attached to the casing by the large flange at the inlet end. The other portion of the nozzle is free to expand.

Another example is illustrated in Fig. 26, which shows a cast steel nozzle box for a marine turbine. The nozzle box is let into a rectangular opening in the cast-iron casing, and its flat sides are supported against the pressure difference by the screwed stays as shown.

Nozzles. For very small turbines in which only one or two nozzles are required, the nozzles are constructed separately, and are often circular in cross-section. This does not produce the best form of jet from the turbine point of view, but it obviously simplifies manufacture.

In larger machines the power required is such as to necessitate a somewhat greater number of nozzles, and a type of construction often adopted is illustrated in Fig. 27. The nozzle casting *a* is segmental in form, and the nozzles proper are formed by " casting-in " the sheet metal vanes *b*, which have been previously bent to the correct shape. The vanes *b* are set in the mould at the correct angle (being embedded in the core) and, when the metal is poured into the mould, the vanes become firmly embedded in the casting. With vanes of uniform thickness *t*, the channels between the vanes form convergent nozzles suitable for expansion down to the critical pressure.

Convergent-divergent nozzles can be constructed in the manner described above by varying the thickness of the vanes. There are various other methods of constructing the first stage nozzles of both the convergent and the convergent-divergent types, but limitations of space prevent any reference to them.

The nozzles in diaphragms are also most generally formed by " casting-in " sheet metal vanes.

Diaphragms. The various " cells " of the turbine are separated by partitions or diaphragms. Almost invariably diaphragms are in halves, although some makers construct them in one piece. While the latter construction gives greater strength to the diaphragm to resist the forces due to pressure difference, it greatly increases the difficulties of erection

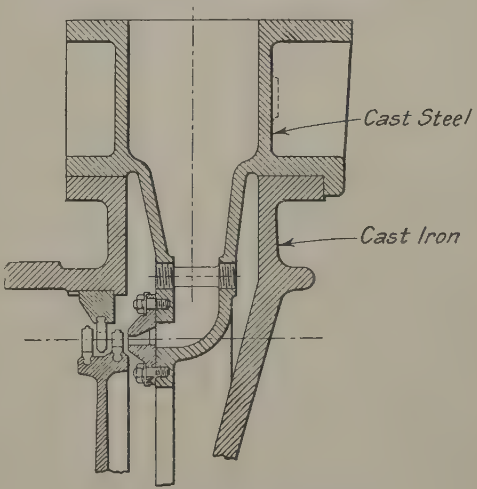


Fig. 26. Cast Steel Nozzle Box for Marine Turbine

and inspection. The diaphragms must be "threaded on" to the shaft alternately with the discs, the complete system of rotor and diaphragms being then lowered into the bottom half of the casing.

When the diaphragms are constructed in halves, the top halves are attached to the top half of the casing, and are raised with it when the turbine is opened up for inspection. The joint between the two halves of the diaphragm is scraped, and there is also a spigot and recess. For temperatures below about 400° F., cast iron is the material usually employed, above that temperature, cast steel. The vanes are of mild steel or nickel steel.

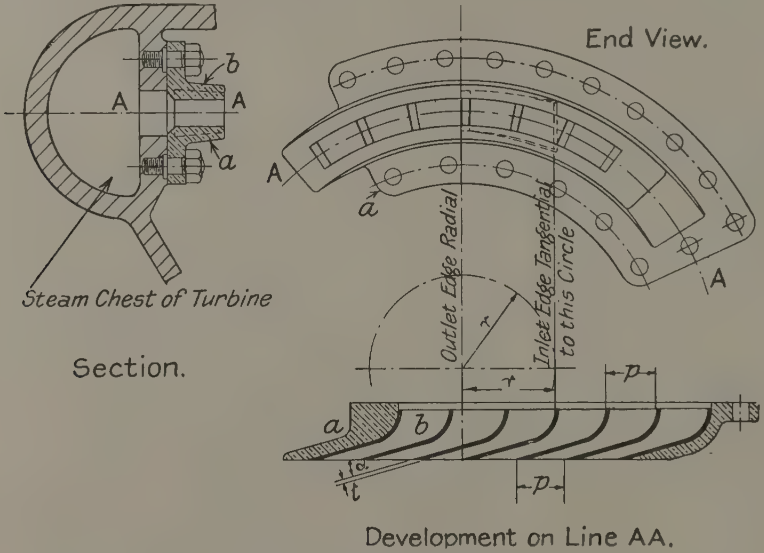


Fig. 27. Impulse Turbine First Stage Nozzles

Leakage of steam between the bores of the diaphragms and the hubs of the rotors is minimized by some form of gland. The commonest form is the well-known *labyrinth gland*, some examples of which are shown in Fig. 28. Perhaps the term "labyrinth" is rather an unfortunate one, because it tends to convey the idea that leakage of the steam is made more difficult owing to the tortuous path which the leaking steam must take. This, however, is not the principle on which labyrinth packing works. The essential feature is that there must be a number of thin strips of metal placed in close juxtaposition to the shaft with, therefore, a very small clearance in which the steam is *throttled*. In other words, the steam first acquires a certain velocity owing to expansion, the kinetic energy so acquired is then reconverted into heat in the "pocket" between the two packing strips, and the steam then undergoes a further expansion while passing through the next clearance space. The number of throttlings is thus equal to the number of packing strips, and the greater the number of such strips the lower the velocity acquired at each throttling, and the smaller the rate of steam leakage.

Referring to Fig. 28, (*a*) shows a simple type of gland in which four strips of gun-metal are fitted directly into the cast-iron diaphragm. The strips are secured by caulking and they are bored out to a fine edge, which is initially allowed to run in contact with the rotor hub. The rotor thus wears its own clearance, and the leakage area is the least possible.

(*b*) shows another variation in which two packing strips are caulked into a separate ring, the ring being dovetailed into the diaphragm. This construction has the advantage over that shown in (*a*) that if, for any reason, the packing strips should be badly damaged, then they may be renewed with much less trouble.

(*c*) shows the type of diaphragm gland which is fitted in multi-stage turbines constructed by the Aktiebolaget de Lavals Angturbin (Swedish de Laval Steam Turbine Company). An S-shaped ring of Siemens-Martin steel is fitted to the bore of the diaphragm, and carries a number of packing strips pitched about $\frac{1}{8}$ in. apart.

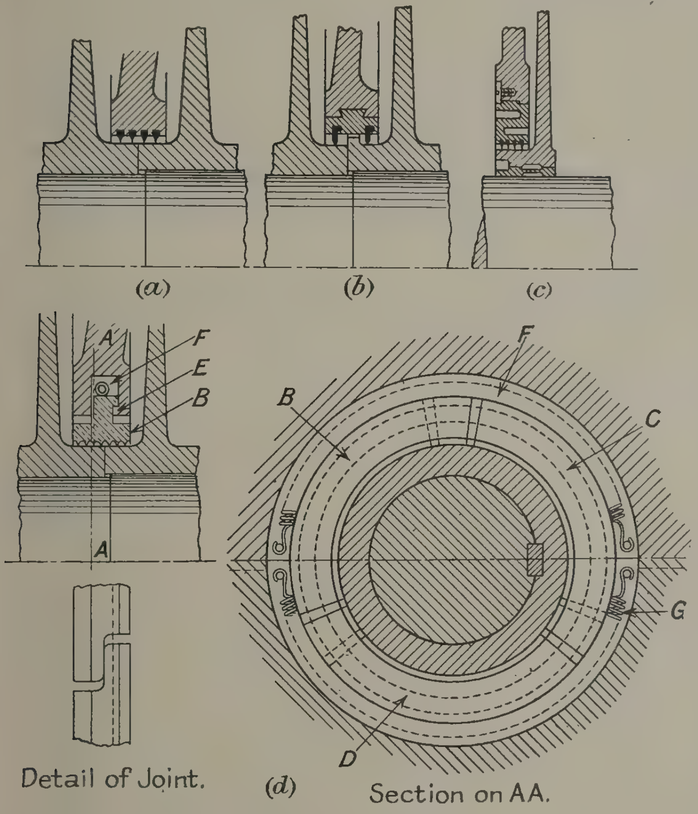


Fig. 28. Details of Diaphragm Glands

A still more elaborate gland is shown in Fig. 28 (*d*). It comprises three segmental gun-metal rings, *B*, *C*, and *D*, provided with a number of fins on their inner surface and held in contact with a lip *E*, machined on the diaphragm, by means of garter springs *F* and *G*. The clearance between the hubs of the wheels and the fins is made about 0·01 to 0·015 in. To prevent leakage between the segments a scarfed joint is used as shown. Should the shaft whip for any reason, the packing rings may be forced outwards against the slight tension of the garter spring without any considerable damage being done to the fins.

Segmental rings of carbon are also used for diaphragm glands. The segments are held up to the shaft either by garter springs or plate springs attached to the back of the segments. Scarfed joints are used as in Fig. 28 (*d*).

External Glands. Present-day constructions of external glands may be classified as follows—

(*a*) Labyrinth glands.
(*b*) Water-sealed glands.
(*c*) Combined labyrinth and water-sealed glands.
(*d*) Carbon packed glands.

A good example of a labyrinth gland suitable for the high pressure end of a steam turbine is shown in Fig. 29. This type of gland is used in turbines constructed by Messrs. Fraser & Chalmer's Engineering Works. It comprises a forged steel sleeve *A* pressed on to the turbine spindle, a forged steel box *C* fitted into the bore of the turbine casing, and three steel rings *E*, *F*, and *G*, which are held in position by the outer casing *D*. A large number of fins are turned on both fixed and moving parts of the gland as shown, and the clearances between the running and stationary parts are made as small as possible. By far the greatest number of constrictions is provided between the inside of the turbine casing and the first pocket. High pressure steam leaking past the fins in *C* and *E* reaches this pocket, and is led by the pipe shown into a stage in the low pressure part of the turbine in which it may do useful work by further expansion to the condenser pressure. The second pocket communicates with the atmosphere by means of a vapour pipe. At very light loads, the pressure in the high pressure part of the turbine casing may, in some cases, be lower than that of the atmosphere, and in order to prevent air leakage into the turbine casing, live steam will then be supplied to the first pocket. Some of this steam will leak into the atmosphere, the remainder will leak through into the turbine.

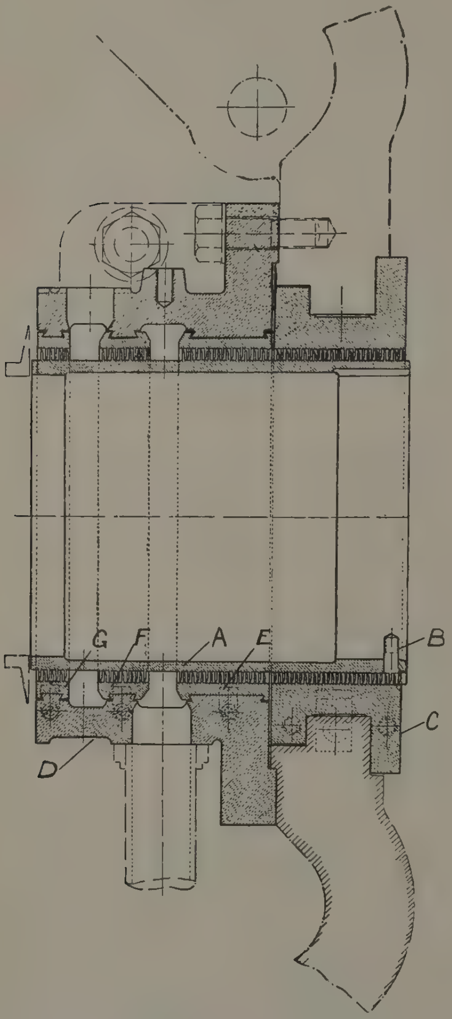


Fig. 29. External Labyrinth Gland for High Pressure Turbine

The amount of steam which leaks through the high pressure gland of a turbine depends upon the design of gland, and upon the size of machine. In machines of moderate output it will not usually exceed 1 or 2 per cent of the steam consumption at full-load.

Fig. 30 is a diagram illustrating the construction and principle of the water seal as used at the low pressure end of steam turbines. An impeller or paddle *A* is attached to the turbine spindle and runs in a circular casing *B*. Water is supplied from an overhead tank to the casing *B*, and on entering the casing the water takes up a rotary motion which is imparted to it by the paddle. Owing to the centrifugal force on the water, the free surfaces at each side tend to remain at the same radial distance from the shaft axis. If, however, the pressure at one side, e.g. the exhaust branch side, of the gland is lower than that at the other side, then the radius r_2 will be less than the radius r_1, and the centrifugal force on the annular ring of water between the radii r_1 and r_2 will just balance the pressure difference $p_1 - p_2$. The radii r_1 and r_2 adjust themselves automatically when the turbine is running so that the pressure in the vortex at the radius r_3 is equal to the head of the water in the supply tank above the gland.

The water seal is probably the best possible type of gland for the low pressure end of the turbine, as atmospheric air cannot possibly leak into the turbine. It is frequently used for sealing off the pocket in high pressure glands, and has the advantage of preserving cool bearings.

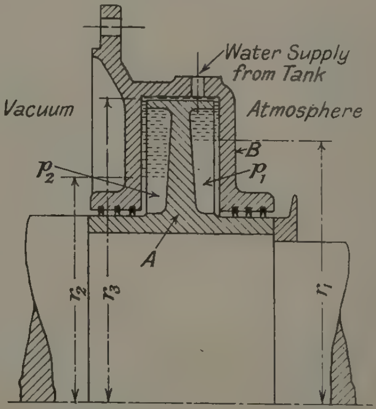


Fig. 30. Diagram of Water-sealed Gland

Carbon rings are very frequently used for external glands. The rings are usually arranged in three segments, and the other arrangements are similar to those for labyrinth glands.

MACHINE TOOLS

By JOSEPH G. HORNER, A.M.I.MECH.E.

LESSON XVII

THE MILLING MACHINES

Early Designs. When the writer was an apprentice there were no milling machines in English shops—those engaged in the manufacture of small arms excepted. Now they are as ubiquitous as any other groups. But the earlier tools were puny weaklings compared with the present, and were confined to two only, the Lincoln, and the pillar and knee designs. Some types, now common, had not come to birth. The cutters were only suitable for fine finishing, they could not rival the single-edged tools for quantity production. The fault of all the early cutters, following the first by Vaucanson in the eighteenth century was the crowding of the finely-pitched teeth which left insufficient space for chips. But that held the germ of present-day vastly developed forms. The first very crude machine was designed for the use of gunmakers in 1818, to finish surfaces prepared by the fitters. No real headway was possible until emery wheels were used for grinding the teeth, for previous to that the temper was drawn and the teeth sharpened with a file.

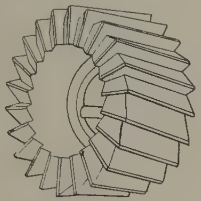


FIG. 116. SPIRAL SHELL MILL

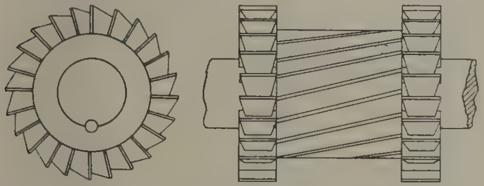


FIG. 117. GANG STRADDLE MILL

In the evolution of types, the Lincoln miller was the earliest. Much later the pillar and knee machines, with universal functions were designed, both having horizontal spindles, and using edge cutters. Then developments and specialization followed rapidly. Vertical spindle machines were introduced, employing both edge and end mills, and then plano-millers, a direct challenge to the planing machines, built mainly after these models, and using both edge and face mills. Attention was directed to increasing the slogging capacities of cutters, in all machines, and to the extension of profile milling by the building up of gang mills to deal with broad surfaces having irregular outlines, combinations of plane, and curved sections.

The Cutters. The advocates of milling in the early days laid emphasis on the obvious facts that a milling tool has multiple edges which cut continuously, and that there is no lost stroke as in the reciprocating tools. Both are real gains, but they are largely discounted, because with increase of edges the depth of cut is correspondingly reduced, and the loss of time consequent on reciprocations in planers has been immensely lessened, and economies have been multiplied relatively to the output produced during a cycle of cutting and return with modern machines.

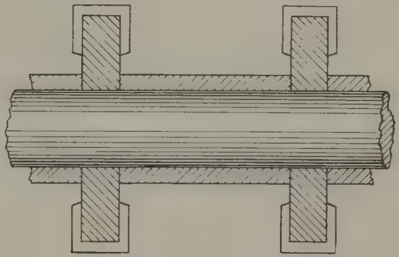


FIG. 118. GANG GROOVING MILLS

All the early cutters were pitched too finely with the object which seemed then of chief importance, that of increasing the number of edges. But the chip clearances were reduced proportionately with choking of the spaces, increased friction, and reduction of the depth of cut. Such tools nibbled only, they could not remove much material, and were liable to spring and produce distortion of the work.

Spacings are now increased, and with other improvements in the shapes of the teeth, the frequent adoption of front rake, the increase in the size of arbors, the better support afforded to them, the growing employment of face cutters, the increased rigidity of machine construction, the use of volumes of cooling liquids, and a more intelligent appreciation of general working conditions, have combined to render the milling cutters of to-day excellent roughing, as well as finishing tools. Generally, special cases apart, the same cutters are used for roughing and finishing, and for all metals and alloys. The special cases are those where very heavy duty is desired, and where work is highly repetitive.

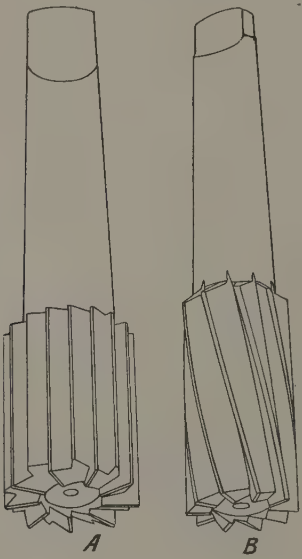


Fig. 119. Straight and Spiral End Mills

It has been demonstrated that coarsely-pitched cutters remove thicker chips, with less expenditure of power than those with fine pitches do. The only merit of finely-pitched tools is, that they leave a smoother surface than the others do. It is not usual to impart front rake to the teeth of the smaller cutters, nor is it adopted much in the formed cutters which are ground on their faces, and used for curved and irregular outlines, and gear teeth. But inserted tooth cutters for heavy duty nearly invariably have angles of front rake, similar and equal to that imparted to the single-edged tools. The angle of clearance is small, say, from 3 to 7 or 8°.

The edge mills are the earliest and still are the largest group of cutters used. The teeth are parallel with the axis of rotation, though in tapered and profile mills the distance of the edges from the axis varies. The teeth run straight with the axis in the smallest tools only. In all above about 1 in. in width they are spiral (strictly helical) in their disposition. This gives a smooth shearing cut, because the edge enters gradually, and the cut proceeds in detail along the edge instead of striking along the whole width abruptly, with shock. The spiral is right- or left-handed. Its amount varies from 10 to 20° or more. When long parallel mills are built up, the spirals are set in reverse directions. For heavy duty the teeth are often nicked in helical paths to break up the chips.

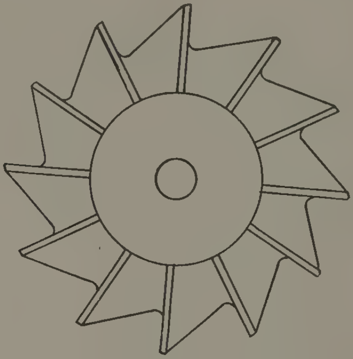


Fig. 120. End View of Mill in Previous Figure

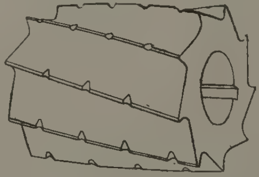


Fig. 121. Coarse-toothed Cutter with Notched Teeth

Fig. 116 shows a spiral shell mill to fit on an arbor with a cylindrical key. Fig. 117 is a gang straddle mill, comprising a central spiral cutter flanked with mills that cut by their edges, so tooling a face and two edges simultaneously. Fig. 118 shows how two gang mills for grooving are mounted on an arbor with a spacing distance piece between, a method that is carried out in a countless variety of forms. Fig. 119 illustrates mills that cut by their ends only when used for facing, but they will also cut with their edges in vertical spindle machines *A* has straight teeth, *B* has negative rake, adopted to prevent the tendency of the tool to dig in under heavy cutting. which would happen if the rake were of the opposite hand. Fig. 120 gives an enlarged end view of the teeth. which are not centre cut. Those that are. have internal teeth which will cut to a depth equal to the length of the end teeth. End mills remove material better than edge mills do, because they are able to penetrate more deeply, and do not spring and yield to pressure.

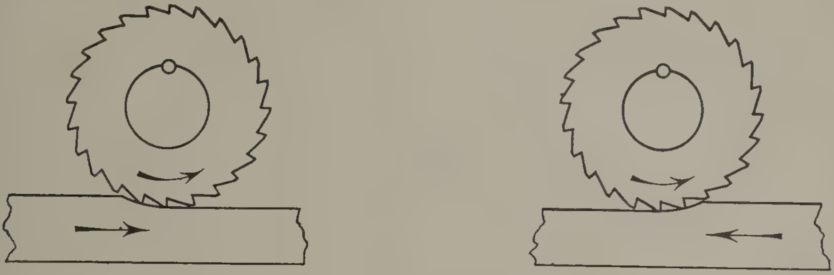


FIG. 122 FIG. 123

INCORRECT AND CORRECT DIRECTION OF FEED OF EDGE MILL

When a difference is made between tools for roughing and finishing, the first are pitched more coarsely to give increased chip space, and the teeth are often notched to break up the chips, Fig. 121. An edge mill must not cut as in Fig. 122, if the surface of the work is rough, because the teeth continually come into contact with scale and dirt, and so become dulled rapidly. The direction of cut must be as in Fig. 123, where the teeth enter on a clean surface. A few examples of edge and face milling are given in Figs. 124, 125, which are self-explanatory. The great utility of these methods over that of using single-edged cutting tools with their numerous readjustments hardly needs emphasizing. These are typical of wider extensions with the inserted tooth mills used in the big plano-millers, cutting profiled sections. Here problems peculiar to the cutters themselves have to be tackled, their production, materials, the retention of their shapes after regrinding, and the manner in which they are supported.

The peripheral speeds of cutters are generally stated as being from two, to two and a half times those of single-edged tools. Using mills of carbon steel, cast iron may be cut at a surface speed of 60 ft. per minute, mild steel at 33 ft., and brass at 125 ft. Tools of high speed steel, which are those now mostly used, will operate at twice these rates. But since speed is closely related to feed, the best results are secured by combining coarse feeding with moderate speeds. Feed is stated either as the advance in inches per minute, or in parts of an inch per revolution of the cutter, or in parts of an inch per single tooth. The net result is estimated by the number of cubic inches removed per minute, which is a measure of the efficiency of speed plus feed. Feed stated in

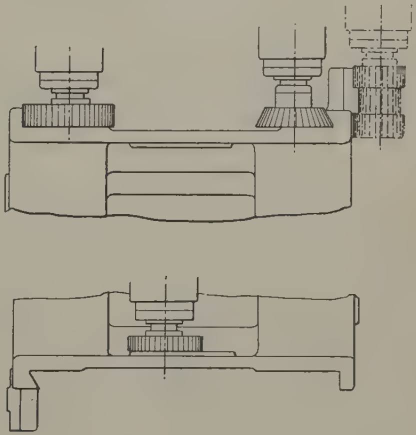


FIG. 124. CUTS WITH VERTICAL SPINDLE

parts of an inch per tooth is more satisfactory than that in inches per minute, or in parts of inches per revolution, because these take no account of the number of teeth in the cutter. A cutter correctly shaped, operating in ideal conditions, will remove large curling chips fairly comparable with those severed with single-edged tools.

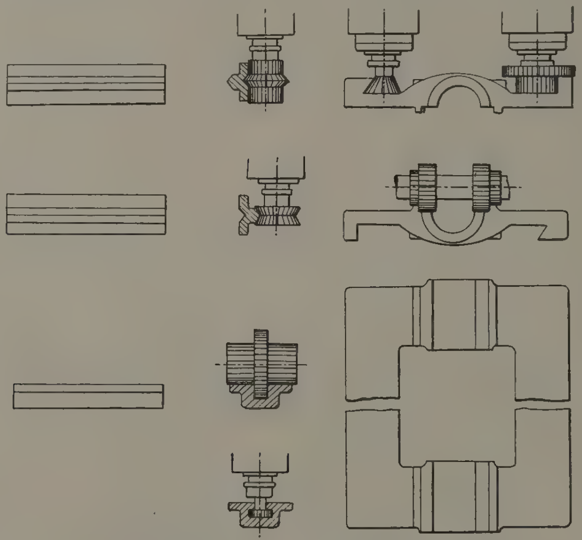


Fig. 125. Examples of Profile Milling

Machines—the Lincoln Miller. This is the lineal descendant of the earliest type, and is still one of the most useful. Its construction renders it suitable for very heavy cutting, the work being supported on a table that slides on a bed beneath the horizontal cutter. This is carried on an arbor driven from a headstock, and supported at the other end on a tailstock. Adjustments for the height of work of different depths have to be made in the arbor, the bearings for which are adjustable on the faces of the uprights. Rigidity of these is increased by tying them together at the top with a stiff bar. The arbor bearings are elevated with screws. A range of speeds is imparted to the arbor, and of feeds to the table through belt cones or gears. The table has usually only a movement along its bed under the cutter, but several of the larger machines have provision for making lateral adjustments relatively to the uprights. The upright that carries the tailstock is generally a fixture, but in some cases it is capable of movement along the bed to increase or diminish the distance available for work.

The value of this machine is that of general utility within its capacity. Single articles, or smaller pieces in tandem can be bolted to the table, or mounted in a fixture, and tooled with single or with gang mills. Its capacity is limited by the width between head and tailstock, and the height from the table to the arbor. Examples will be shown in the next lesson.

FITTING AND ERECTING

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XI

THE ERECTION OF ENGINES

Although the work of the fitter has been reduced because of the greater care that is now devoted to the machining of the component parts of engines, the tasks of the erector, regarded apart from those of the mere assembler, have not been materially affected. And every class of engine presents its own peculiar problems and difficulties, and the scope of the subject is as wide as the variations in their forms. It will be necessary, therefore, to concentrate attention on the leading facts that apply to all engine erections.

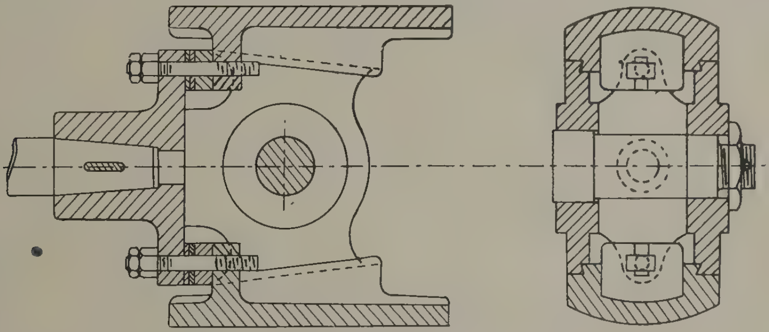


Fig. 73. Crosshead for Bored Guide

Essential Facts. One important distinction from the point of view of the erector is that between vertical and horizontal designs. Another is that of single-cylinder engines, and of compounds, either in tandem or in parallel, and again that of single engines, and of engines duplicated side by side on the same foundation, or bedplate. There are also many differences between the stationary and the locomotive types, and those derived from the latter, the road rollers and agricultural engines. Oil and gas engines are groups apart in most of their details. Yet all engines have many features in common, such as alignments, and right angular relations. Neither can be checked and corrected with the small instruments of the bench, but geometrical devices—straining lines, plumb-bobs, levels, and templets are requisitioned.

Very slight departures from correct relations result in friction, in hard working of the engines, and rapid heating up of journals and bearings, with probable scoring, and permanent damage to the parts.

Stationary Engines

For all these, a levelled bedding for the base, or the bed, is necessary if the work of erection is to be satisfactory. The base for a vertical engine, or the bed for a horizontal is received from the machine shop with its working faces tooled, but the bottom is as roughly cast. The casting is levelled and wedged up on blocking until the faced portions show truly horizontal when tested with a spirit-level. Lengths of balk or quartering are suitable for engines of moderate dimensions, but for the heaviest a solid permanent bed is preferable to the dirt floor, and metal wedges for levelling. The spirit-level should be mounted on a parallel straight-edge to bridge over adjacent facings of the same height, and the level be turned end for end as a check on its possible inaccuracy. The facings then become the datum, whence cylinders and shaft can be tested for parallelism and heights of centres.

Horizontal Engines. In the erection of these, the commencement can be made with the cylinder, or the crankshaft—generally with the former. The essential is, that they shall occupy strictly linear, and right-angular relations to each other, for if they do not, then very slight departures from accuracy will set up friction, wear, and heating between the crank journals and their bearings, or the piston and the cylinder bore, or the crosshead and its guides. The usual method of aligning is to bridge the rear end of the cylinder with wood or metal, find the exact centre from the bore, and drill a hole there to receive a straining line of fine wire. For repetitive work, a permanent guide may be registered into the bell mouthing of the cylinder and clamped against the flange. The line, passing through the cylinder, is continued beyond the crank shaft and, being packed up there into a horizontal position, is pulled taut with a heavy weight suspended at each end. Although it is impossible to strain a long line perfectly straight, centres must be widely separated for this to have a serious influence if the line is heavily loaded, as piano wire can be. An indicator gauge or a templet may be used to check the sag, and packing may be used if necessary midway between the cylinder and the

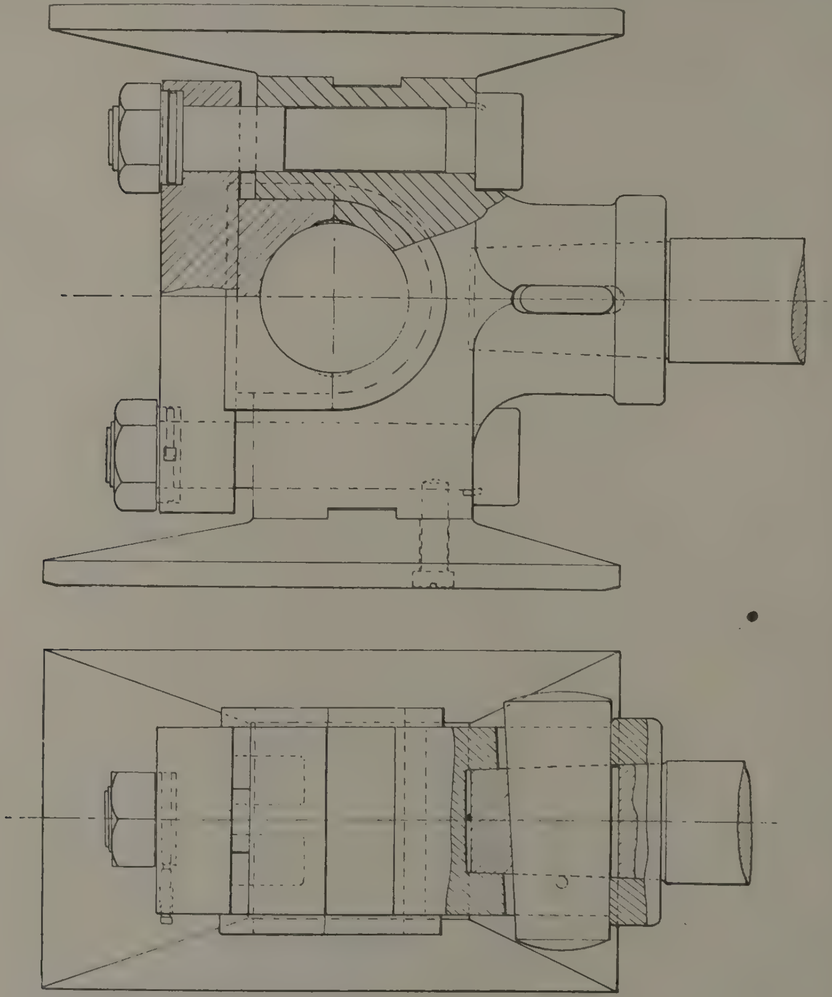


Fig. 74. Crosshead for Flat Guides

crankshaft bearings. The line will be straight laterally.

From the line set centrally at the rear of the cylinder the front of the cylinder is aligned, and the crosshead guide, or the guide bars and the centre of the crankshaft. The detailed methods will differ with the construction of the engine. In some designs the cylinder and the guides are cast solidly with the bed. More often they are separate castings bolted in place, which permits of making adjustments. Crankshaft bearings are generally cast with the bed, but in the larger engines they may be separate fittings. When parts are cast solidly, the only corrections and adjustments that can be made are those in the brasses for the crankshaft journals. The accurate relations of parts cast together divide responsibility between the marker-off and the machinist. In such work, the horizontal boring machines with tables that can be reset at 90° are of much value. since cylinders and crankshaft bearings can be tooled with a reasonable probability that they will be at exact right angles.

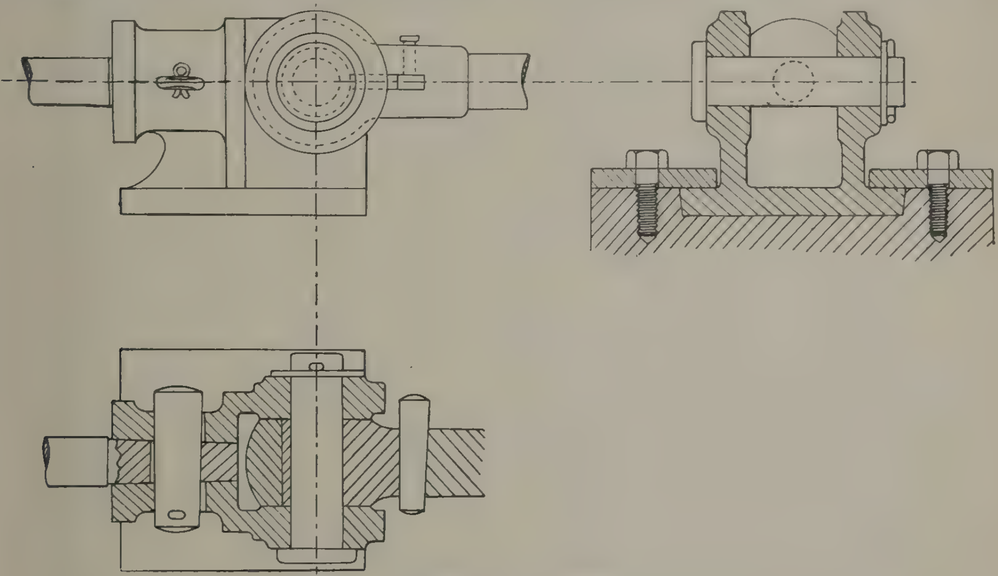


FIG. 75. CROSSHEAD FOR A SINGLE FLAT GUIDE

Separate cylinders may be bolted down on the bed, or the crosshead guides may be cast with the bed, and the cylinder be bolted at the back end—the overhanging type. In the first case, adjustments of the cylinder through its feet are readily made, in the second, adjustments can be made through the end flange, or the face against which it is bolted. This may sometimes give less trouble than adjustments at the crankshaft end. A lack of alignment in the cylinder bore will be multiplied five or six times at the crankshaft.

For the front end of the cylinder and for the guides, measurement is taken from the wire, round the bores, using internal calipers, or a rod gauge, handled delicately, not to disturb the wire, which if it is sufficiently strained will indicate lack of alignments. If corrections are found necessary they must be made with file or scraper. Holes are finally reamed for the fastening bolts.

Bored guides, having arc sections above and below, with which the crosshead makes sliding contact are most common. Fig. 73 shows one design of this type of crosshead. Fig. 74 slides between flat faces, Figs. 75 and 76 slide on a single flat face. In another, an early form, the crosshead is pierced with a gudgeon at right angles, carrying at its ends slipper blocks that move between upper and lower guide bars having plane surfaces. In these cases, the lower face on which the crosshead slides has to be checked from the straining line, and also transversely to be sure that it is horizontal—

not higher on one side than on the other. To check these, blocks are planed to a thickness equalling that from the centre of the cylinder to the lower face, and these being moved about along and across are checked with a spirit-level, when any departure from truth in either direction is easily detected. Then if there are top guide bars, these are fitted parallel with those below.

This work has to be repeated when engines are compounded with cylinders in tandem or in parallel, with cylinders cast together, or separately, and with an air pump at the rear, and when valve. casings for piston valves are included. When, as in the largest engines, the main elements are carried on separate bases of masonry or concrete, these have to be levelled in relation to each other, and the engine bases often have to be grouted before erection is commenced.

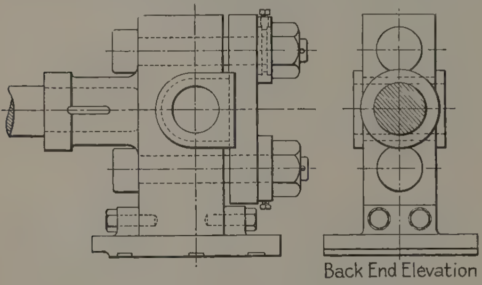


FIG. 76. CROSSHEAD FOR A SINGLE FLAT GUIDE

It is essential that the crankshaft shall be at exact right angles with the axis of the engine, and that, when a dip crank is used the crankpin shall be central laterally with the axis. If a disc crank is fitted, the shaft bearing alone need have attention. To check the right angular relation with a square against the strained line to another line strained across the shaft bearings is possible if the square is supported on blocking a trifle higher than the line and a sight taken down the edges. The line for the shaft is laid in shallow notches cut in strips of wood that bridge each shaft bearing, and crossing the longitudinal line at the centre of the crankpin. A more reliable method is to make a templet, a tee-square, the blade of which is laid along one side of the cylinder bore, the other over the notches cut in the pieces that bridge the shaft-bearings, or against the strained wire. If the templet is true it will show any departure from the right-angled relations of both bearings. Another way, equally reliable if the cylinder has been accurately faced in relation to the bore, is to make a templet, one portion to lie against the front flange of the cylinder, the other to check the centre of the crankshaft.

The course of procedure now varies, being controlled by the amount of work that has been done on the bearings. If the tooling has followed the lining off, with hand-setting on machines, errors may occur in each. The tooling is that of the seatings for the brasses, with the fitting of the caps, followed by the boring and facing of the brasses. If error is likely to develop, it is well to defer the boring of the brasses to a later stage. After the seatings are bored, the shoulders for the caps planed or milled, and the stud holes drilled and tapped and the studs inserted, the only adjustments that are possible are those on the brasses. These can be bored out of centre, and packings inserted to accommodate the correct position of the crankshaft. The centres of the brasses will be obtained from the cross wire in one direction, and in the vertical with a surface gauge or a level on blocking, and the circle of the bore struck from the point of intersection. After the boring, the flanges will be faced to gauges.

There are besides these cardinal features, other matters of scarcely less moment, which become apparent at later stages, and in working, viz., the character of the fitting of piston rings, crossheads, connecting rods, valve rods, valves, stuffing boxes, packings, the consideration of which belongs equally to the vertical engines—the subject of the next lesson.

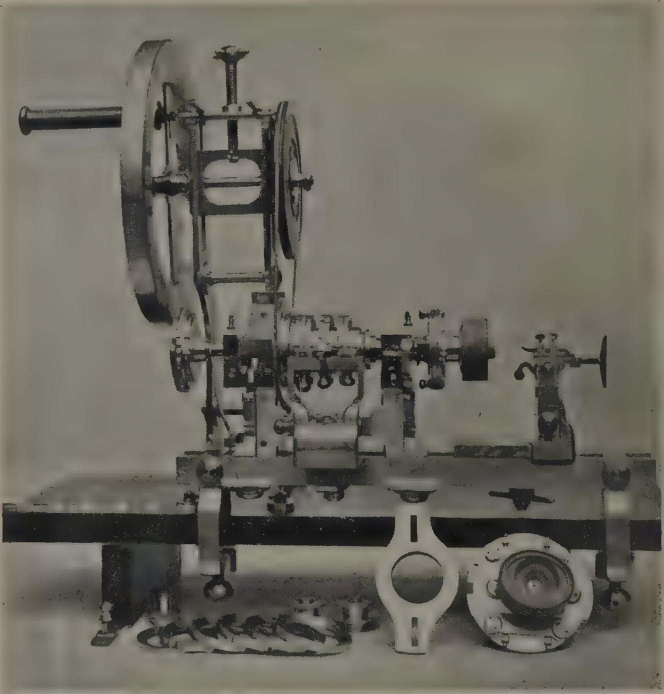


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FIG. 40. FRENCH "ROSE-ENGINE" LATHE, *c.* 1740

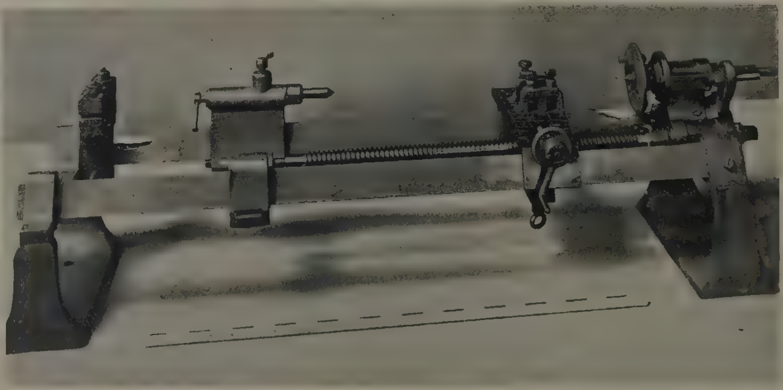

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FIG. 41. MAUDSLAY'S SCREW CUTTING LATHE, *c.* 1800

(See Article on "Notable Events in Engineering History," page 1079)

MATHEMATICS FOR ENGINEERS

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LESSON XXII

DIFFERENTIATION

At the end of the last lesson we had found that for a falling body, when the distance gone (s ft.) is connected with the time (t sec.) by the formula $s = 16t^2$, the instantaneous speed at the end of 2 sec. was 64 ft./sec. The result was obtained by arithmetically calculating the average speeds over decreasing intervals, and seeing to what these tended as the intervals were reduced. By using a symbol for the small interval of time (δt), we can find the result with less labour, one piece of algebra taking the place of a number of pieces of arithmetic.

From the formula we have, when $t = 2$, $s = 64$. Now following the motion for an interval δt sec., and letting δs represent the distance travelled in the interval, the total distance, $64 + \delta s$, will have been travelled in the time $2 + \delta t$ sec. So, by the formula,

$$64 + \delta s = 16(2 + \delta t)^2 = 64 + 64\,\delta t + 16\delta t^2$$

Consequently,

$$\delta s = 64\delta t + 16\delta t^2$$

To find the average speed in the interval, we must divide the distance δs by the time δt, obtaining

$$\frac{\delta s}{\delta t} = 64 + 16\delta t$$

Now we see that as δt becomes less, the value of $\frac{\delta s}{\delta t}$ approaches 64, or in symbols,

$$\text{as } \delta t \to 0,\ \frac{\delta s}{\delta t} \to 64$$

or, since the value that the average velocity approaches must be the instantaneous velocity we are seeking,

$$\frac{ds}{dt} = 64$$

At this point, before proceeding farther, the reader should try to find, by the above method, the values of $\frac{ds}{dt}$ when $t = 3$, 5, and 10 (or any other values of t he fancies). For the values given, the results should be 96, 160, and 320.

After doing this, and seeing that exactly the same process is used each time, it will be evident that by keeping a symbol for time (instead of separate numerical values, 2, 3, 5, and 10) we should be able to obtain a *formula* for $\frac{ds}{dt}$ from which it can be found for *any* value of t.

We have $s = 16t^2$ . . . . . (1)

At time δt sec. later ($t + \delta t$) the total distance gone ($s + \delta s$) must be by the formula,

$$\begin{aligned} s + \delta s &= 16(t + \delta t)^2 \\ &= 16t^2 + 32t.\,\delta t + 16\delta t^2 \qquad (2) \end{aligned}$$

Subtracting (1) from (2), we find

$$\delta s = 32t.\,\delta t + 16\delta t^2$$

Dividing by δt to find the average speed,

$$\frac{\delta s}{\delta t} = 32t + 16\delta t$$

From this we see that as $\delta t \to 0$, $\frac{\delta s}{\delta t} \to 32t$, so

that $$\underline{\frac{ds}{dt} = 32t}$$

a result which will be found to agree with all the numerical results already obtained.

Again, starting with the formula $y = 3x^2$, let us find the formula for $\frac{dy}{dx}$ in the same way.

We have $y = 3x^2$

so $y + \delta y = 3(x + \delta x)^2 = 3x^2 + 6x\,.\,\delta x + 3\delta x^2$

Subtracting, $\delta y = 6x\,.\,\delta x + 3\delta x^2$

therefore $$\frac{\delta y}{\delta x} = 6x + 3\delta x$$

Hence, as $\delta x \to 0$, $\frac{\delta y}{\delta x} \to 6x$, or $\underline{\frac{dy}{dx} = 6x}$

This process of finding rates of change $\left(\text{i.e. such quantities as } \frac{ds}{dt} \text{ and } \frac{dy}{dx}\right)$ is called differentiation, and the rates themselves are called differential coefficients. Before giving the general rule for differentiating powers, we will work a few more examples—

1. $y = 2x^4$

$$y + \delta y = 2(x + \delta x)^4$$
$$= 2x^4 + 8x^3\delta x + 12x^2\delta x^2 + 8x\,.\,\delta x^3 + 2\delta x^4$$

expanding the power of $(x + \delta x)$ by the binomial theorem (see Lesson XII). Subtracting, we find

$$\delta y = 8x^3\delta x + 12x^2\delta x^2 + 8x.\delta x^3 + 2\delta x^4$$

so $$\frac{\delta y}{\delta x} = 8x^3 + 12x^2\delta x + 8x.\delta x^2 + 2\delta x^3$$

As $\delta x \to 0$, $\frac{\delta y}{\delta x} \to 8x^3$, all the other terms tending to zero, so $\frac{dy}{dx} = 8x^3$

In this example every term has been worked out in full, but it must be evident that it is not necessary to know more than the first two of the terms of the expansion in order to find the value of $\frac{dy}{dx}$, so that, *provided we know that they will not affect* $\frac{dy}{dx}$, we may save ourselves the labour of calculating and writing them. Any term in the expansion which contains a power of δx higher than the first will after division by δx, still contain δx, and will consequently tend to zero as δx does. The next instance will show the minimum of writing and calculating.

2. $y = 3x^7$

$$y + \delta y = 3(x + \delta x)^7$$
$$= 3x^7 + 21x^6\delta x + \text{terms in } \delta x^2\text{, etc.}$$

Subtracting,

$$\delta y = 21x^6\delta x + \text{terms in } \delta x^2\text{, etc.}$$

Dividing,

$$\frac{\delta y}{\delta x} = 21x^6 + \text{terms in } \delta x\text{, etc.}$$

As $\delta x \to 0$, $\frac{\delta y}{\delta x} \to 21x^6$, so $\frac{dy}{dx} = 21x^6$

3. $y = \frac{1}{x}$

$$y + \delta y = \frac{1}{x + \delta x}$$

Here there is no simple expansion, but we can still obtain δy by subtraction,

$$\delta y = \frac{1}{x + \delta x} - \frac{1}{x} = \frac{x - (x + \delta x)}{x(x + \delta x)} = -\frac{\delta x}{x(x + \delta x)}$$

on using the common denominator and performing the subtraction. Dividing, we have

$$\frac{\delta y}{\delta x} = \frac{-1}{x(x + \delta x)}$$

As $\delta x \to 0$, $\frac{\delta y}{\delta x} \to -\frac{1}{x^2}$, so $\frac{dy}{dx} = -\frac{1}{x^2}$.

The meaning of the minus sign in this example is important. Since $\frac{dy}{dx}$ means the rate of *increase* of y with respect to x, a negative rate of increase might be expected to mean a rate of *decrease*—and it does, for as x increases, it is evident that $\frac{1}{x}$ decreases.

Rule for Powers. All the above instances are particular cases of a general rule for differentiating powers which we now proceed to obtain.

If $y = x^n$

$$y + \delta y = (x + \delta x)^n$$
$$= x^n + nx^{n-1}\,\delta x + \text{terms in } \delta x^2\text{, etc.}$$

Subtracting,

$$\delta y = nx^{n-1}\,\delta x + \text{terms in } \delta x^2\text{, etc.}$$

Dividing,

$$\frac{\delta y}{\delta x} = nx^{n-1} + \text{terms in } \delta x\text{, etc.}$$

As $\delta x \to 0$, $\frac{\delta y}{\delta x} \to nx^{n-1}$, so $\frac{dy}{dx} = nx^{n-1}$

This is the rule sought. Expressed in words,

To differentiate any power, **multiply by the index, and reduce the index by 1.**

If we had started with ax^n instead of x^n, every quantity on the right of the $=$ would have been multiplied by a, and so we should have found that $\frac{dy}{dx} = nax^{n-1}$—the rule being again, multiply by the index and reduce the index by 1. All the examples already worked agree with this formula—

$s = 16t^2$, $\frac{ds}{dt} = 2 \times 16t^1 = 32t$ (as above)

$y = 3x^2$, $\frac{dy}{dx} = 2 \times 3x^1 = 6x$ (,,)

$y = 2x^4$, $\frac{dy}{dx} = 4 \times 2x^3 = 8x^3$ (,,)

$y = 3x^7$, $\frac{dy}{dx} = 7 \times 3x^6 = 21x^6$ (,,)

$y = \frac{1}{x} = x^{-1}$, $\frac{dy}{dx} = -1 \times x^{-2} = -\frac{1}{x^2}$ (,,) In the last of these we see that the rule is still true for negative indices, and that if we have to differentiate a fraction with x in the denominator, we can express the fraction by negative indices and then differentiate by rule. The same is true for roots, which can be expressed by fractional indices (though we are not going to prove it at length). Thus, if $y = 5\sqrt{x}$, and we want $\frac{dy}{dx}$, we first express the root by the index $\frac{1}{2}$, that is, $y = 5x^{\frac{1}{2}}$, and then the rule gives $\frac{dy}{dx} = \frac{1}{2} \times 5x^{-\frac{1}{2}} = 2{\cdot}5x^{-\frac{1}{2}} = \frac{2{\cdot}5}{\sqrt{x}}\left(\text{since } x^{-\frac{1}{2}} = \frac{1}{x^{\frac{1}{2}}} = \frac{1}{\sqrt{x}}\right)$.

As further instances—

$$y = 15x^6,\ \frac{dy}{dx} = 6 \times 15x^5 = 90x^5$$

$$y = \frac{4}{x^3} = 4x^{-3},\ \frac{dy}{dx} = -3 \times 4x^{-4} = -12x^{-4} = -\frac{12}{x^4}$$

$$p = Cv^{-1.4},\ \frac{dp}{dv} = -1{\cdot}4 \times Cv^{-2.4} = -1{\cdot}4Cv^{-2.4}$$

$$s = \sqrt[3]{t} = t^{\frac{1}{3}},\ \frac{ds}{dt} = \tfrac{1}{3}t^{-\frac{2}{3}} = \frac{1}{3\sqrt[3]{t^2}}$$

and so on.

There are two simple cases of differential coefficients which can be obtained by very simple reasoning. The first is when y is proportional to x, i.e. $y = ax$, where a is a constant. Here, since y is always a times x, the change in y must always be a times the change in x, and so $\frac{dy}{dx} = a$. This result may also be obtained from the formula, by putting in the index 1. For, by the formula, if $y = ax^1$, $\frac{dy}{dx} = 1 \times ax^0 = a$, since $x^0 = 1$. Again, if y is constant, it does not change, so that its rate of change is zero. Again, remembering that $x^0 = 1$, we may write $y = a$ as $y = ax^0$, and the formula will give $\frac{dy}{dx} = 0 \times ax^{-1} = 0$.

Sums and Differences. If y contains two or more terms, connected by + or – signs, the value of $\frac{dy}{dx}$ can be found by differentiating term by term. For, in the case where y is the sum of two terms, it should be clear that the rate of change of y is the sum of the rates of change of the two terms, since the actual change in y is always the sum of the actual changes in the two terms. Exactly similar reasoning shows that if y is the difference of two terms, then the value of $\frac{dy}{dx}$ is the difference of the differential coefficients of the two terms. This reasoning can evidently be extended to any number of terms. Thus, if $y = 2x^3 - 3x^6 + 4x^2$, we have $\frac{dy}{dx} = 6x^2 - 18x^5 + 8x$. We cannot, however, say that the rate of change of a product is the product of the rates of change of the factors—the rule for products will be given later. At present, if we can work out the product algebraically, we can differentiate after doing so. Thus, if $y = (2x + 1)\ (x - 2) = 2x^2 - 3x - 2$, then $\frac{dy}{dx} = 4x - 3$. Quotients are more complicated, unless the denominator is simply a power of x. If $y = \frac{x^2 + 3x - 2}{x} = x + 3 - 2x^{-1}$ (upon division and expressing $\frac{2}{x}$ by an index) $\frac{dy}{dx} = 1 + 2x^{-2} = 1 + \frac{2}{x^2}$.

Other Notations. In many cases we wish to write the differential coefficients (or, more accurately, instructions to find it) without denoting the quantity by a single letter. Considering the notation $\frac{dy}{dx}$ as meaning "the rate of change of y with respect to x," we see that the $\frac{d}{dx}$ can be taken to mean the "rate of change of." Thus the rate of change of $15x^6$ will often be written $\frac{d}{dx}(15x^6)$. In this way we have

$$\frac{d}{dx}(15x^6) = 90\,x^5,\ \frac{d}{dv}\ (Cv^{-1.4}) = -1{\cdot}4Cv^{-2.4}, \text{ etc.}$$

Sir Isaac Newton, one of the inventors of the calculus, used a different notation. To denote the rate of change of a quantity, he placed a dot on top. Thus, he would have written $\dot{s}$ for $\frac{ds}{dt}$. This notation has the advantage of being concise, and is still used in some books on mechanics to denote *time* rates of change. Dashes are also used upon occasion to denote differentiation, thus, y' is sometimes written for $\frac{dy}{dx}$.

Meanings of Differential Coefficients. We already know that if s represents distance travelled, and t represents time, then $\frac{ds}{dt}$ represents speed, or velocity. In the same way, if θ represents angle turned through, and t represents time, then $\frac{d\theta}{dt}$ will represent rate of change of angle with respect to time, or rate of turning —which is angular velocity. Again, if v represents velocity, then $\frac{dv}{dt}$ will represent rate of change of velocity, or acceleration. If ω represents angular velocity, then, similarly, $\frac{d\omega}{dt}$ will represent angular acceleration. In Newton's notation, angular velocity would be denoted by $\dot{\theta}$, acceleration by $\dot{v}$, and angular acceleration by $\dot{\omega}$. There are also many other quantities that can be represented by differential coefficients. Coefficient of linear expansion, for instance. If l is the length of a bar at temperature θ, the coefficient of expansion is found by dividing the increase of length per unit length by the rise of temperature producing it. If δl is the increase of length, and $\delta\theta$ the rise of temperature, the coefficient of expansion is given by $\frac{\delta l}{l} \div \delta\theta$ or $\frac{1}{l}\frac{dl}{d\theta}$, in the limit when $\delta\theta \to 0$. Similarly, if H is the quantity of heat required to raise the temperature of 1 lb. of a substance from 0° C. to θ° C., then the specific heat is given by $\frac{dH}{d\theta}$. Again, if p and v denote pressure and volume of a gas, and θ temperature, the pressure and volume coefficients are given by $\frac{1}{p}\frac{dp}{d\theta}$ and $\frac{1}{v}\frac{dv}{d\theta}$ respectively. In this case, too, the coefficient of elasticity is given by the increase of pressure divided by the *de*crease of volume per unit volume produced, so $E = \delta p \div -\frac{\delta v}{v}$ or, in the limit, $E = -v\frac{dp}{dv}$.

In the above, we have seen that acceleration $= \frac{dv}{dt}$, and that $v = \frac{ds}{dt}$. Thus, acceleration $= \frac{d}{dt}\left(\frac{ds}{dt}\right)$. This is usually written $\frac{d^2s}{dt^2}$, and read as the second differential of s with respect to t. If we have a formula for s, to find $\frac{d^2s}{dt^2}$ we differentiate s twice in succession with respect to t. Similarly, $\frac{d^3y}{dx^3}$ means that we are to differentiate y three times with respect to x. Thus if

$$y = 5x^4 - 3x^2 + 8x$$

$$\frac{dy}{dx} = 20x^3 - 6x + 8$$

$$\frac{d^2y}{dx^2} = 60x^2 - 6$$

$$\frac{d^3y}{dx^3} = 120x.$$

EXERCISE No. 35

1. Find $\frac{dy}{dx}$ for the following values of y—

(*a*) $5x^6 - 3x^3 + 2x + 7$.

(*b*) $\frac{3}{x^2} - \frac{2}{x^3} + \frac{1}{x}$.

(*c*) $\sqrt{x^3} - 5\sqrt{x}$.

(*d*) $6x^{1\cdot 2} - 3x^{0\cdot 4} + 8x^{-0\cdot 4}$.

2. Expand $(x + 3)^3$ by the binomial theorem, and differentiate your expansion. Show that the result can be written $3(x + 3)^2$. Show similarly that $\frac{d}{dx}(2x + 1)^4 = 2 \times 4(2x + 1)^3$, and that $\frac{d}{dx}(ax + b)^n = an\,(ax + b)^{n-1}$.

3. If $pv^n = C$ (a constant) show that $v\frac{dp}{dv} = -np$.

4. If $V = \frac{JL}{144T}\frac{dT}{dp}$, $T = 140p^{\frac{1}{4}} + 465$, $J = 778$, $L = 1{,}431 - \frac{1}{2}T$, find V when $p = 81$.

Answers to Exercise No. 34

1. 11·46° (5·73° on each side).
2. Approximation gives 0·8573.
3. Formula given makes sin 30° = 0·4997 instead of 0·5000.

MACHINE CONSTRUCTION AND DRAWING

By G. WOOD, A.M.I.MECH.E.

LESSON XXI

Drawing Office Practice. Having, in previous lessons, dealt with various technical aspects of machine construction and drawing, we may proceed to consider the art relative to drawing office practice. Most orders for machinery are the outcome of competitive tendering. This means that, at the time of obtaining the order, much of the preliminary design is already done. Consequently, we may, as a rule, split the drawing office share of the work connected with an order into two parts, the tender drawing and the working drawing.

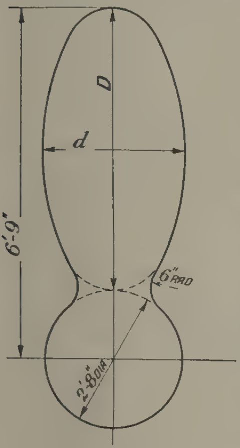

FIG. 1

For an example, we may take a drawing office method of dealing with the following inquiry.

One ship's propeller, right handed, having the following particulars—

Diameter of propeller . . = 13 ft. 6 in.
Pitch of propeller . . = 14 ft.
Projected area of four propeller blades . . . . = 49 sq. ft.
Diameter of propeller shaft . = 14 in.

Power transmitted by set of steam turbines = 4,000 at a propeller speed of 120 rev. per min. Total thrust required from propeller at above power = 40,000 lb.

MATERIAL. Manganese bronze, tensile strength, not to be less than 33 tons per sq. in., with an elongation of 15 per cent in 2 in. and elastic limit 17 tons per sq. in.

MACHINING. To be finished machined to templates supplied by owners. Surface of blades to be filed or ground smooth and bright.

Before proceeding with the design we shall be advised to survey the limits set by practical considerations. One limit set by economy is weight, and a diametrically opposite limit is imposed by the requirements of reliability. As a compromise we have to produce the cheapest design consistent with reasonable strength. If price was the sole factor determining the choice of tender, makers would be tempted to submit designs having a minimum margin of strength. Reliability, however, is very important, and the successful quotation will be one combining a moderate margin of strength with a minimum price. We shall be advised, therefore, to follow accepted practice in deciding the strength of the design, and attempt economy in weight and labour where such economy does not interfere with the required strength.

Commencing our preliminary design with the shaft to which the propeller is to be attached we find that, in transmitting the power to the blades, a boss having a diameter 22 per cent greater than the shaft would provide a section of strength equal to that of the shaft. Such a size of boss would, however, unduly limit the width of blade at the root where the greatest strain occurs. We find from practice that a diameter equal to about $2\frac{1}{4}$ shaft diameters is required to provide the necessary landing for the blades.

We may now proceed, as in Fig. 1, with a circle representing the boss. The term *projected area*, referred to in the inquiry, means the total area of the blades projected on to a plane normal to the axis of rotation. In producing an outline of the blade, Fig. 1, we have to aim at a figure having an area equal to $49 \div 4 = 12\frac{1}{4}$ sq. ft. The outline of blade suitable for the above speed of revolution will be elliptical, and so we may make a rough estimate of axes required to give the necessary area. Tentatively, aiming at an ellipse of requisite area having a major axis equal to the distance between the boss and blade tip, we have, where D and d represent the major and minor axes of the ellipse respectively,

$$\text{Area} = \cdot 7854\,Dd \text{ approximate.}$$

$$d = \frac{\text{area}}{\cdot 7854 D} = \frac{12\cdot 25 \times 144}{\cdot 7854 \times 65} = 34\cdot 55 \text{ in.}$$

Proceeding, we may produce an approximate ellipse with the method suggested in Lesson VIII. and connect the same to the boss as in Fig. 1. As a check on the above we may quickly find the area of the blade by means of a planimeter and, if the area differs materially from the specified $12\frac{1}{4}$ sq. ft., the blade width can be corrected *pro rata.*

Having found a suitable outline for the blade we may now consider the question of suitable thickness. Analysing the manner in which the propeller blade converts the power transmitted by the shaft into ship propulsion we find that the blade rotating in the sea meets resistances in two directions, as in Fig. 2. The one set of resisting forces act normal to the direction of rotation, and the other set parallel to the axis of rotation. The magnitudes of the forces vary with the distance from the axis of rotation and, consequently, in considering the blade as a loaded cantilever we have an example of an irregularly loaded beam. In estimating we have to remember that the prospect of obtaining the order is possible rather than probable, and we cannot, therefore, afford much time in fixing the tender design.

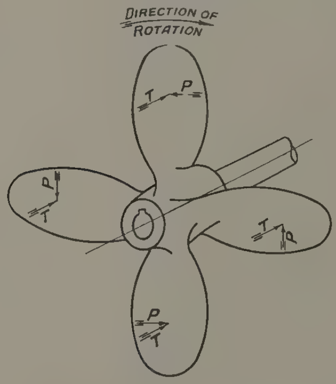

FIG. 2

Approximate Nature of Estimating. The above offers a good example of the approximate method which frequently has to be adopted in fixing the minimum dimensions for an estimate design.

Approximately, the blade under working conditions consists of a cantilever loaded, as shown in Fig. 3, i.e. with the load varying directly as the distance from O, the axis of rotation. The resulting bending moment will be equal to the sum of the moments of the various forces at the different radii, and will be equal to the product of the total load and the distance x of the centre of pressure from the point O.

For a beam of uniform width and the above manner of loading the position of the centre of pressure would be given by

$$x = \frac{2(R^3 - r^3)}{3(R^2 - r^2)},$$ and this approximation may safely be made in the case of a blade having an elliptical outline.

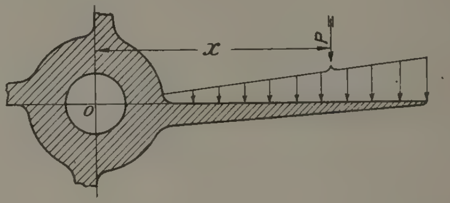

FIG. 3

We may now find the magnitude of the resultant forces P and T acting on each blade.

Let

h.p. = horse-power transmitted by propeller shaft.
N = speed of propeller in revolutions per minute.
T_q = twisting moment transmitted through propeller boss in pounds-inches.
x = distance of centre of pressure from axis of rotation in inches.
P = total tangential load on each blade in pounds.
T = total thrust on each blade parallel to axis of rotation in pounds.

Then

$$T_q = \frac{33{,}000.\ \text{h.p.}\ 12}{2\pi N} = \frac{33{,}000 \times 4000 \times 12}{2\pi \times 120}$$

$$= 2{,}100{,}000 \text{ lb.-in.}$$

and

$$x = \frac{2(81^3 - 16^3)}{3(81^2 - 16^2)} = 55{\cdot}75 \text{ in.}$$

$$P = \frac{T_q}{4x} = \frac{2{,}100{,}000}{4 \times 55{\cdot}75} = 9418 \text{ lb.}$$

$$T = \frac{40{,}000}{4} = 10{,}000 \text{ lb.}$$

We may now proceed to find the maximum bending moment on the root section of the blade, due to the components of the above two forces, taken normal to the blade at the root.

Let

θ = angle made by blade at root to the plane of rotation in degrees.

R = resultant force acting on blade in a direction normal to the blade at the root in pounds.

Then

$$R = P \sin \theta + T \cos \theta.$$
$$R = 9418 \sin 59{\cdot}1^\circ + 10{,}000 \cos 59^\circ{\cdot}1$$
$$R = 8063 + 5135 = 13{,}198 \text{ lb.},$$

and the maximum bending moment (B.M.) occurring at the blade root $= R(x - r)$.

$$\begin{aligned} \text{B.M.} &= 13{,}198(55{\cdot}75 - 16) \\ &= 13{,}198 \times 39{\cdot}75 = 526{,}200 \text{ lb.-in.} \\ &= 234 \text{ tons-in.} \end{aligned}$$

Before proceeding with the outline of the blade section at the root we may consider the method of finding the lie of the blade at various radii. Drawing lines OX, OY at right angles, Fig. 4, and transferring the blade outline from Fig. 1, draw a line ab parallel to OX at a distance of *pitch* $\div 2\pi$ from same. Next draw arcs of varying radii cutting the lines OX and OY. The radius of the blade at the root $= r$ and cuts the line OX at c. Then a line drawn through c and a will give the angle of helix at radius r. Proceeding, projections from the points d on the blade intersecting the line ac at e and e defines the width of blade at the root.

In deciding the outline of blade section we observe that for a clockwise direction of rotation the working face of the blade lies to the right of the line ac. This side of the section will be straight, the opposite side being curved as shown in Fig. 4. The moment of resistance R.M. offered by such a section in a line normal to ac is approximately $\dfrac{b_1 d_1^2}{10} f$ where

b_1 = width of blade in inches.

d_1 = thickness of blade in inches.

f = stress induced in tons per square inch.

B.M. = bending moment on section in tons-inches.

R.M. = resisting moment of section in tons-inches.

Then R.M. = B.M.

$$\frac{b_1 d_1^2}{10} f = 234$$

A practical value for $f = 2{\cdot}3$

$$\frac{32 d_1^2}{10} \times 2{\cdot}3 = 234$$

$$d_1 = \left(\frac{234 \times 10}{32 \times 2{\cdot}3}\right)^{\frac{1}{2}} = 5{\cdot}64 \text{ in.}$$

say, 5·75 in.

It often happens that forces, acting upon a machine detail, are of comparatively small magnitude and not so quickly determined as the greater forces. Without the experience to provide a confident judgment of what may be ignored it is advisable to make an approximate and generous estimate of the force in question. As an example of such approximation we may take the effect of the centrifugal forces acting in the above propeller blade when rotating at the specified maximum speed of revolution.

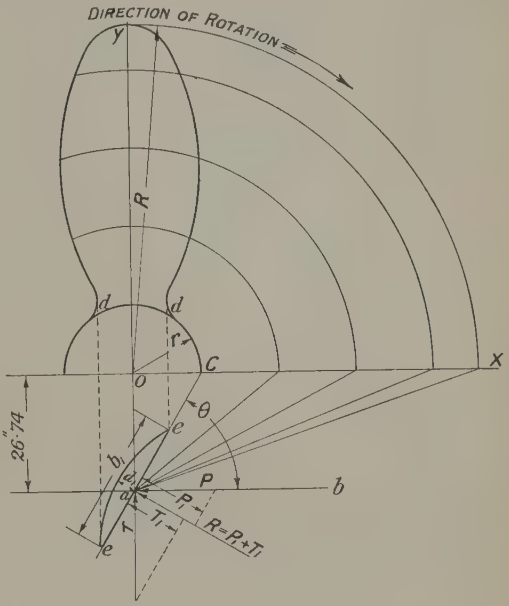

FIG. 4

Let

W = weight of blade in pounds.

w = speed of propeller in radians per second.

r = distance of blade centroid from axis of rotation in feet.
A = sectional area of blade at root in square inches.
F = centrifugal force in pounds.
f_1 = tensile stress induced at blade root in pounds per square inch.

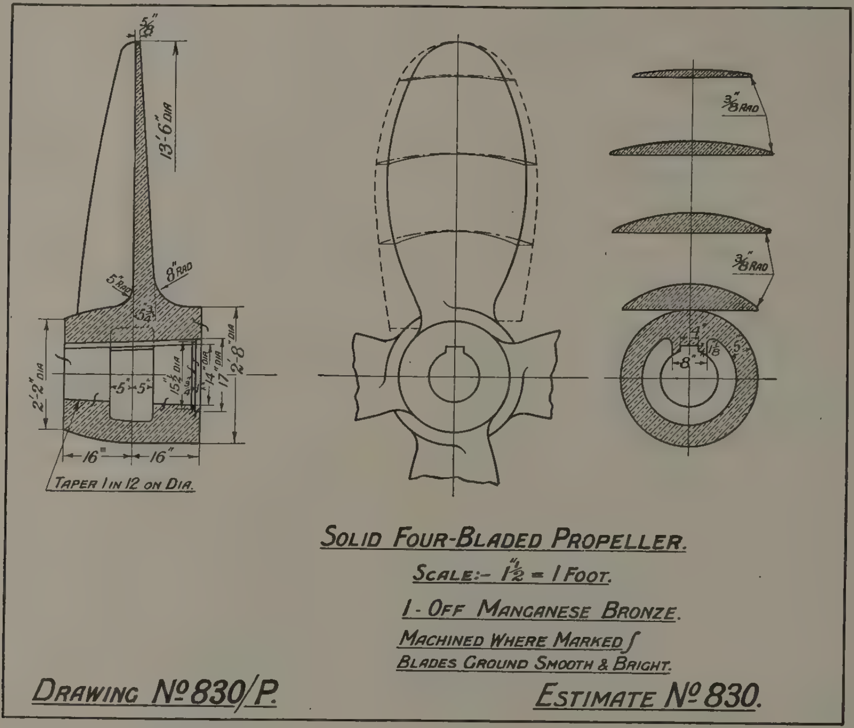

FIG. 5

$$\text{Then } w = \frac{2\pi \times 120}{60} = 12{\cdot}57$$

$$F = \frac{Ww^2r}{g} = \frac{1520 \times 12{\cdot}57^2 \times 3{\cdot}1}{32{\cdot}2}$$

$$= 23{,}100 \text{ lb.}$$

$$f_1 = \frac{F}{A} = \frac{23{,}100}{123} = 188 \text{ lb. per sq. in.}$$

$$= {\cdot}084 \text{ tons per sq. in.}$$

The above calculation, although so approximate, is sufficient to show that the stress, due to the action of the centrifugal forces acting on the blade, is sufficiently small (compared with the bending stresses) to be neglected.

It will be observed that the outline of blade section at the root determines the minimum length of boss required to support the blade. With the above information we may proceed with the tender drawing as in Fig. 5. The tender drawing is not sufficiently informative to be considered a working drawing, and is intended for quotation purposes only. In the next lesson we will proceed with the above tender, and consider the method of obtaining the weight for the purpose of fixing a price.

EXERCISE

Make a tender drawing for an inquiry similar to the above with the following exceptions—

Diameter of propeller	= 12 ft.
Pitch of propeller	= 12 ft.
Projected area of propeller	= 38 sq. ft.
Diameter of propeller shaft	= 12¾ in.
Thrust on propeller shaft	= 32,000 lb.
Power transmitted by propeller shaft	= 2,800 s.h.p.
Speed of propeller shaft	= 125 r.p.m.

ELECTRO-TECHNICS FOR MECHANICAL STUDENTS

By A. P. Young, O.B.E., M.I.E.E.

LESSON X

TYPES OF D.C. GENERATORS—THEIR CHARACTERISTICS AND APPLICATIONS

Electric generators are classified according to the field winding connections, there being three types—

(*a*) Shunt-wound generator.
(*b*) Series-wound generator.
(*c*) Compound-wound generator.

(*a*) **Shunt-wound Generator.** The connections of the field winding of a shunt-wound generator are shown in Fig. 63, from which it will be seen that the field winding is connected directly across the brushes. A rheostat is usually inserted in the field circuit for controlling the field current and thereby the output of the generator.

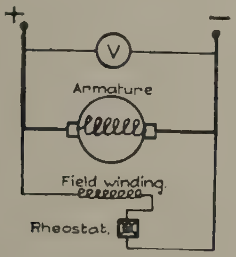

Fig. 63. Connection Diagram for Shunt-wound Generator

The terminal voltage of a shunt-wound generator is a maximum on open-circuit and diminishes as the load increases, in the manner shown in Fig. 64. This drop in voltage is due to—

1. The IR drop in the armature.
2. The demagnetizing effect of the armature current.
3. The reduction in field current, consequent upon the lower terminal voltage caused by (1) and (2).

Shunt-wound generators are suitable for steady load conditions, and where the voltage variation is not of great importance. They are generally used for exciting the field windings of alternators and rotary converters. Also, they are used for battery charging purposes.

(*b*) **Series-wound Generator.** In a series-wound generator, the field winding is connected in series with the armature winding, as illustrated in Fig. 65. As the armature current increases, so also does the field current, and this results in a voltage curve having a rising characteristic as shown in Fig. 66. If the field current is such as to saturate the iron, any further increase in load will cause a decrease in voltage. This is mainly due to an increase in the armature demagnetizing component, without a corresponding increase in field strength.

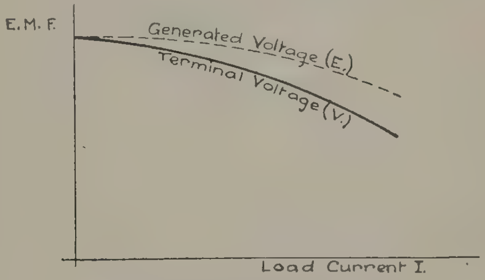

Fig. 64. Characteristic Curves for Shunt Generator

In order to vary the field strength of a series-wound machine, it is usual to connect an adjustable resistance or *diverter* in parallel with the field winding.

This type of machine finds but very limited application, as it is quite unsuited for dealing with variable loads as ordinarily met with in practice. It is essentially a constant current machine and was used extensively, some years ago, for supplying current for arc lamps on the *series system.*

(*c*) **Compound-wound Generator.** Compound-wound generators having both shunt and series windings are employed for general purposes demanding a constant voltage under varying conditions of load. The connections of a compound wound generator are shown in Fig. 67. The shunt winding maintains the terminal voltage at no-load, and the series winding provides the additional flux to compensate for the drop in voltage on load due to armature resistance and reaction. It will be evident that by suitably proportioning the series winding in relation to the shunt winding, the voltage characteristics of the generator can, to some extent, be controlled.

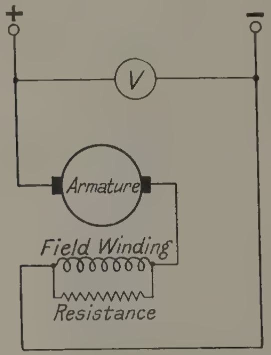

Fig. 65. Connection Diagram for Series-wound Generator

Fig. 68 shows characteristic voltage curves for a compound-wound generator. When the series winding is arranged just to compensate for the voltage drop as a result of the armature resistance and reaction, the machine is said to be *flat-compounded*, and a practically straight line characteristic OB is obtained. If the series winding is insufficient, the voltage curve will " droop " at the higher loads (see curve OC), and the machine will be *under-compounded*. On the other hand, should the series turns be greater than is required for flat-compounding, the machine is said to be *over-compounded*, and the voltage will rise as the load increases, as indicated by curve OA. When a generator is required to supply power at some distance, *over-compounding* is necessary to compensate for the increased line voltage drop at the higher loads. In this way, the voltage at the load terminals can be maintained practically constant.

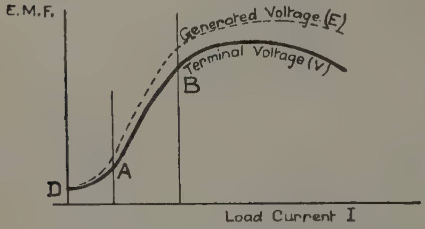

Fig. 66. Characteristic Curves for Series Generator

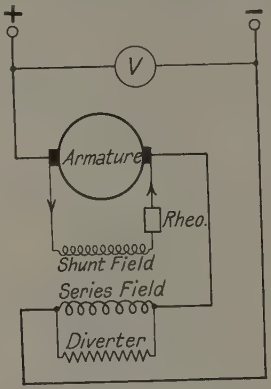

Fig. 67. Connection Diagram for Compound-wound Generator

Losses and Efficiency of a D.C. Generator. The losses in a D.C. generator may be classified under two groups—

Constant Losses	*Variable Losses*
Iron or core loss.	Armature I^2R.
Shunt I^2R.	Series- and commutating pole I^2R.
Bearing friction and windage.	Brush I^2R.
Commutator friction.	

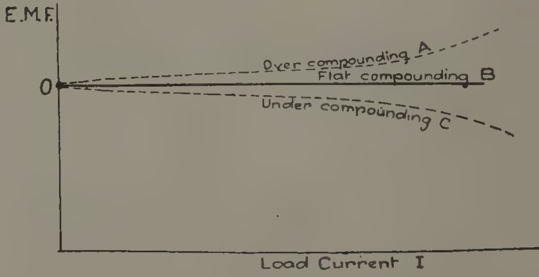

Fig. 68. Characteristic Curves for Compound Generator

The commercial efficiency is given by the formula—

$$\text{Commercial efficiency} = \frac{\text{electrical output (h.p.)}}{\text{mechanical input (h.p.)}}.$$

If *horse-power* (h.p.) is required to drive a generator giving *I amps.* at *V volts*, then the commercial efficiency is—

$$\text{Commercial efficiency} = \frac{I \times V}{746 \text{ h.p.}} \quad . \quad (25)$$

The electrical efficiency is expressed by the equation—

$$\text{Electrical efficiency} = \frac{Wo}{Wo + W_A + W_F} \quad . \quad . \quad (26)$$

Where,

Wo = output in watts $(I \times V)$

W_A = Watts consumed in armature winding and,

W_F = Watts consumed in field winding.

DIRECT CURRENT MOTOR

Principle of Operation. In the previous lesson we have described how an electric current is induced in an armature as a result of rotation in a stationary magnetic field; that is, by expending mechanical energy we produce electrical energy. As in other cases of energy transformation we can reverse the process, and by so doing convert electrical energy into mechanical energy. This is done in the electric motor, and any direct current generator can be run as a motor by supplying current to the armature from an external source. The D.C. motor is, therefore, identical in construction to the D.C. generator, but in order to limit the first rush of current on starting, a suitable rheostat is inserted in the armature circuit.

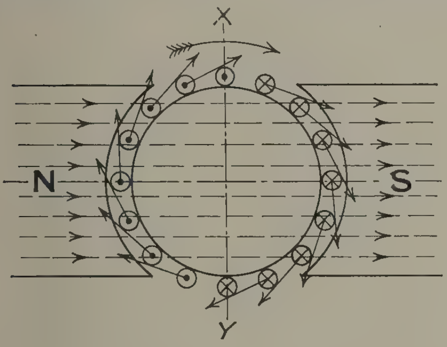

FIG. 69

In Lesson VII it was explained that when a conductor carrying current is placed in a magnetic field with its axis at right angles to the flow of the magnetic lines, a force is exerted on the conductor in a direction at right angles to the magnetic lines of force. The force urging a single straight conductor of l cm. and carrying I amp., across a magnetic field of strength B lines per sq. cm., is given by the formula—

$$\text{Force} = F = \frac{I \times B \times l}{10} \text{ dynes} \quad . \quad (27)$$

When current flows in the armature windings of an electric motor, each conductor is acted upon by a force, as indicated in Fig. 69. The resultant effect is that the armature experiences a torque causing it to accelerate until a limiting speed is reached. During the short period required for the armature to run up to speed, the starting resistance is gradually cut out of circuit until ultimately the armature is connected directly across the supply mains.

It is evident that as the armature is rotating in a magnetic field, an E.M.F. must be induced in it which, according to Lenz's Law, will oppose the flow of current giving rise to the motion. In other words, the *induced E.M.F.* resulting from *rotation* and known as *the back E.M.F.* (*E*) *opposes the applied E.M.F.* (*V*), and the *armature current* will *result from the difference between the two E.M.F.'s*, i.e. $(V - E)$.

If the resistance of the armature winding is R, the current for steady conditions is expressed by the equation—

$$I = \frac{V - E}{R} \quad . \quad . \quad . \quad . \quad . \quad (28)$$

The back E.M.F. (E) is proportional to speed (n), and the flux per pole (Φ), and is given by the formula—

$$E = \frac{(SX)}{y} \times \frac{\Phi n}{10^8} \quad . \quad . \quad . \quad . \quad (29)$$

Where,

S = No. of armature conductors.

X = No. of poles.

y = No. of parallel paths in armature

$= \frac{1}{X}$ for lap winding

$= \frac{1}{2}$ for wave winding

Since the armature current is proportional to $(V - E)$, it is necessary to insert resistance in the armature circuit at starting when E is zero, in order to limit the current. This resistance is gradually cut out of circuit as the speed—and in proportion thereto E—increases.

When the motor is running light, the torque necessary to maintain continuous rotation is small, and correspondingly, the armature current (I). If we assume that the armature voltage drop ($I \times R$) is very small and negligible in comparison with the supply voltage (V), we then have—

$$V = E = \text{constant} \times (\text{flux} \times \text{speed}) \qquad (30)$$

This relationship is substantially correct for the no-load condition, and can be taken as a rough approximation for other loads. Thus, if the supply voltage (V), or the flux per pole Φ, be varied, then the speed will automatically adjust itself to give the relationship expressed by equation (30). Any increase in voltage (V) will result in an increase in speed, whilst if V be kept constant and the flux varied by adjusting the field current, *an increase in speed will result from a decrease in flux, and vice versa, any increase in flux will be followed by a corresponding decrease in speed, since the product* ($\Phi \times n$) *will remain substantially constant.*

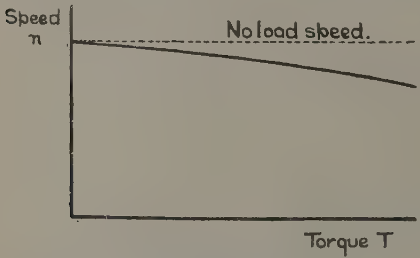

FIG. 70. TYPICAL SPEED CURVE FOR SHUNT MOTOR

If the load on the motor be increased, then the armature current must also increase, and equation (28) tells us that this condition can only be fulfilled by a reduction in the back E.M.F. (E). *Any increase in load will therefore be accompanied by a reduction in speed* sufficient to allow the armature current to rise to produce the torque required by the new load conditions. By analysing equation (28), we can deduce an expression for the value of the armature torque corresponding to any load condition. Rewriting this equation, we get—

$$V = E + RI.$$

Multiplying throughout by I we have—

$$VI = EI + RI^2 \quad . \quad . \quad . \quad . \qquad (31)$$

Now VI is the power in watts supplied to the motor, and RI^2 the power dissipated in the form of heat in the armature, so that EI represents the energy expended in doing mechanical work plus that expended in overcoming frictional and iron losses. Assuming that—

T = armature torque in lb. ft.
n = speed in rev. per sec.

then

$$\text{Power} = \frac{2\pi n T}{550}\text{ h.p.} = \frac{746 \times 2\pi}{550} \times nT \text{ watts.}$$

Equating this to the product ($E\,I$), we get—

$$T = \frac{550}{746 \times 2\pi} \times \frac{EI}{n} \quad . \quad . \quad . \qquad (32)$$

Substituting the value given for E in equation (29) we have—

$$T = \frac{550}{746 \times 2\pi} \times \frac{SX}{y} \times \frac{\Phi I}{10^8}\text{ lb. ft.}$$

or,

$$T = \frac{0{\cdot}1174\, SX}{10^8 y} (\Phi I)\text{ lb. ft} \quad . \quad . \qquad (33)$$

TYPES OF D.C. MOTORS

THEIR CHARACTERISTICS AND APPLICATIONS

Just as in the case of electric generators, there are three distinct types of D.C. motors, viz.—

(*a*) Shunt-wound motor.
(*b*) Series-wound motor.
(*c*) Compound-wound motor.

The change in speed for varying conditions of load, or torque, will be different in each case.

(*a*) **Shunt-wound Motor.** The field connections are the same as for the shunt-wound generator, so that the voltage across the field circuit will be the supply voltage. Neglecting armature reaction, the flux Φ will remain practically constant at all loads, so that any increase in load will cause a gradual decrease in speed. The characteristic speed curve is given in Fig. 70. For constant supply voltage, *the drop in speed is almost directly proportional to the torque*, as will be evident from equation (28), bearing in mind that E and I are substantially proportional to the speed and torque respectively.

The maximum drop in speed is but a relatively small percentage of the normal running speed, and for practical purposes, the shunt-wound motor is a constant speed machine. It therefore finds wide application for driving machine tools and other mechanical appliances requiring fairly constant speed operation.

(*b*) **Series-wound Motor.** As the field winding is in series with the armature, any increase in load will result in a rapid increase in field strength. We have seen from equation (30) that the back E.M.F. (E) remains *approximately constant* under all load conditions, and as this factor is proportional to the product ($\Phi \times n$) we can say that this product is also approximately constant for all loads. *Any increase in load is, therefore, followed by a rapid increase in flux, and a correspondingly rapid drop in speed during such increase.* The characteristic curve shown in Fig. 71 consequently differs widely from that of the shunt-wound motor. When the load is small, the speed is extremely high, and if connected directly across the supply mains on *no-load*, the speed would be *dangerously high.* Although the reduction in speed with increase in load is very great, the series motor has a distinct advantage in its capability to exert a very large torque at starting. Since the torque is proportional to Φ and I (see equation (33)), it follows that *as the flux increases with the current, the torque must increase at a still greater rate.*

Series motors are used extensively on electric cranes, tramcars, and electric railways, where a large starting torque is required. The characteristic curve (Fig. 71) indicates that when heavily loaded—as represented by the portion of the curve between A and B—the speed variation is not excessive, and under these conditions a series motor has somewhat similar torque speed characteristics to a shunt motor. This is due to the iron in the poles becoming saturated, after which the field remains substantially constant.

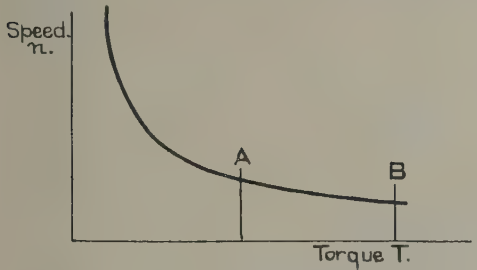

FIG. 71. TYPICAL SPEED CURVE FOR SERIES MOTOR

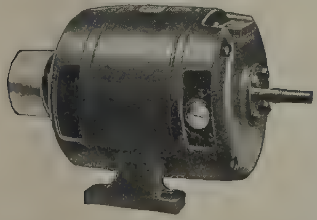

FIG. 72. A TYPICAL FRACTIONAL HORSE-POWER MOTOR
(General Electric Co.)

(*c*) **Compound-wound Motor.** In this type we have a combination of the shunt- and series-wound motors, the field connections usually being arranged so that the effects of the two field windings are cumulative. For special purposes, where a constant speed is required under all load conditions, the series winding opposes the shunt winding, and is arranged to weaken the main field by an amount just sufficient to prevent a drop in speed, due to the increase in armature current with increase in load. This arrangement is known as *differential* winding.

Cumulative winding gives a mean between the characteristics of series- and shunt-wound motors, i.e. a large starting torque and a speed at no-load which is not unduly high. These motors are employed for fluctuating loads, such as are met with when operating rolling mills, stamping presses, and lifts.

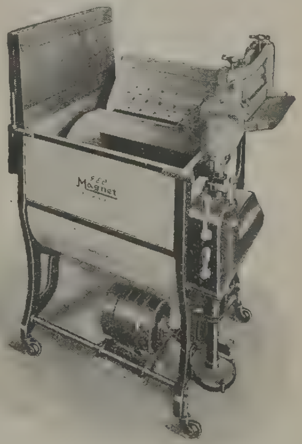

FIG. 73. APPLICATION OF A FRACTIONAL HORSE-POWER MOTOR TO AN ELECTRIC WASHING MACHINE
(General Electric Co.)

Fractional horse-power motors of the compound-wound type, as shown in Fig. 72, are now extensively used for both industrial and domestic purposes. The field of application of this class of motor will, undoubtedly, extend enormously during the next few years, and already it has done much to bring electric power into the home. These small motors find innumerable applications in the engineering, textile, printing, boot, and many other industries, for driving small machines of every description. A domestic application of the fractional horse-power motor is shown in Fig. 73, whilst Fig. 74 will serve to illustrate its application in the engineering industry.

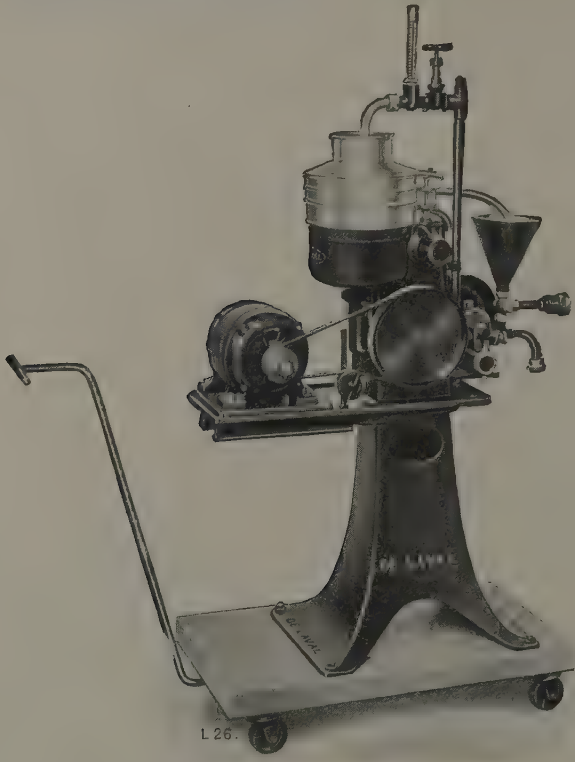

Fig. 74. Application of a Fractional Horse-power Motor to an Oil Separator
(Alfa-Laval Co.)

Typical characteristic curves for compound motors are shown in Fig. 75.

Efficiency of an Electric Motor. The commercial efficiency is calculated as follows—

$$\text{Efficiency} = \frac{\text{mechanical output in watts}}{\text{electrical input in watts}}$$

$$= \frac{Ho \times 746}{V \times I} \quad . \quad . \quad . \quad (34)$$

Where,
Ho = brake horse-power
V = supply voltage
I = current taken by motor (amp.).

Efficiencies range from 95 per cent for motors of 500 h.p. and over, to 85 per cent for 5 h.p. sizes. Motors of lower output will have an efficiency somewhat less than 85 per cent.

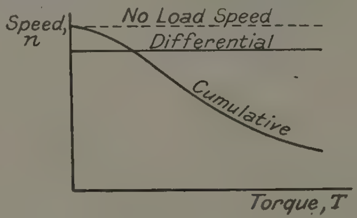

Fig. 75. Typical Speed Curves for Compound Motors

NOTABLE EVENTS IN ENGINEERING HISTORY

By E. A. Forward, A.R.C.S., M.I.Mech.E.

LESSON X

MACHINE TOOLS

With hand tools, the form and accuracy of the work produced depends entirely on the skill of the operator. When using soft materials, such as wood or ivory, a satisfactory result could be obtained without excessive labour, but when iron came into general use the labour involved in cutting it, and the imperfect result obtained, led to the introduction of machine tools, the essential feature of which is that the cutting tool is fixed in a holder and guided by mechanical means.

There are two distinct features in machine

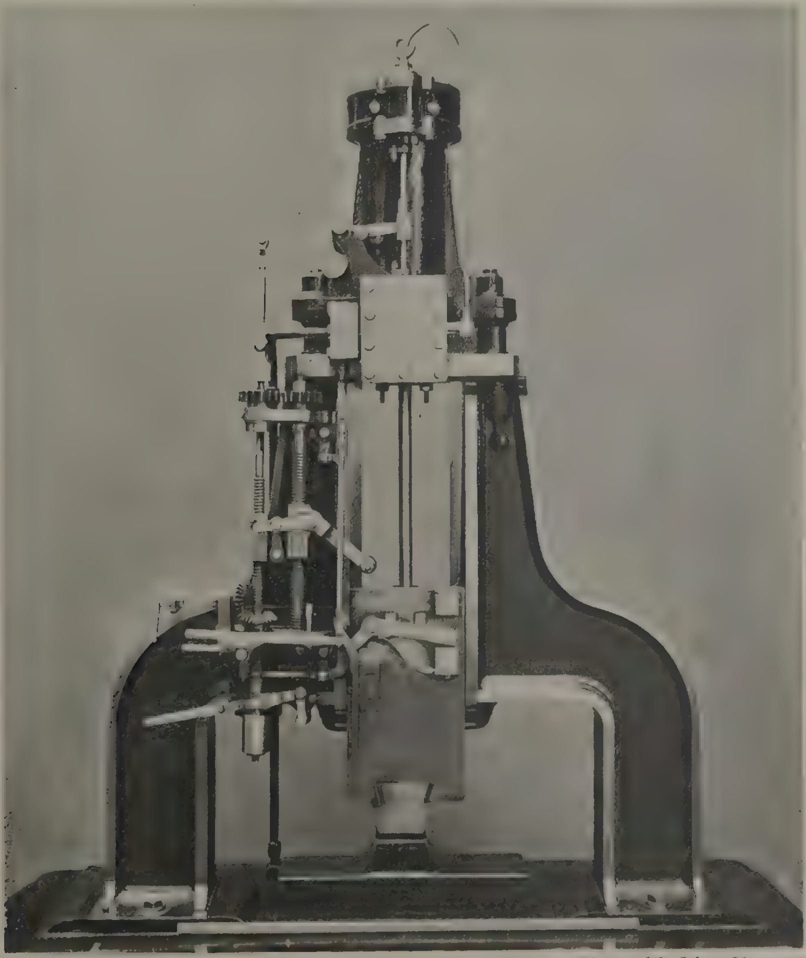

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Fig. 42. Nasmyth's Steam Hammer, 1839

tools that have each promoted the rapid development which took place during the nineteenth century. First, they do their work almost entirely by the forces of nature, without animal labour, and so are without limit of size or endurance, and second, the speed and accuracy with which they attain any desired result is due to the copying principle now to be found in all finishing tools.

For the production of accurate machine tools, the use of true planes is necessary. A tool guided by two planes inclined to one another describes a straight line, and in all these tools such lines and planes are being copied on to the work.

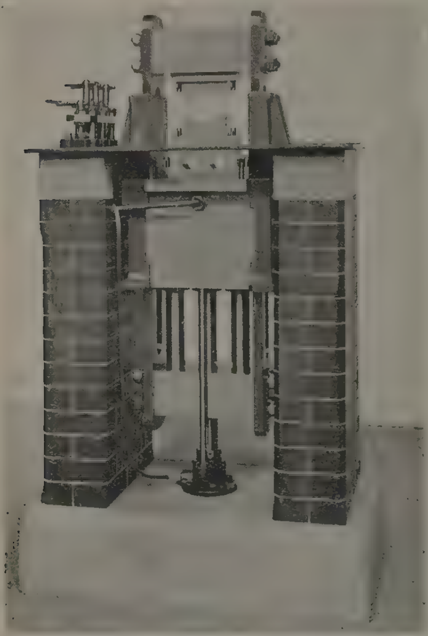

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Fig. 43. Hydraulic Forging Press

Metal working machine tools perform the desired alteration in the shape of the material, either by forcing it to flow into the required form as in forging, rolling, and stamping, or the result is obtained by cutting from a larger mass those portions not required.

In hammering tools, the copying principle is hardly applied, but in rolling and stamping the great saving in labour and time that it effects is fully seen.

The hammer was probably the first tool to which power driving was applied, and it was one of the first to be driven by Watt's steam-engine. The greatest advance, however, was made by James Nasmyth, in 1839, when he invented his direct-acting steam hammer (Fig. 42). Forging in dies under a hammer greatly facilitates the production of accurate work, and special mechanical hammers for this work were introduced by Hall, in America, in 1827. Instead of forging by hammer blows, it is now becoming increasingly general to work the heated metal by the steady pressure exerted by a hydraulic press (Fig. 43). This process was commenced by Bessemer in 1856, and continued by Whitworth in 1861, whilst Messrs. Tannett and Walker did much to further its early progress.

The use of grooved rolls in a rolling mill was practically introduced by Henry Cort in 1783.

Lathes are tools of great antiquity, but they remained small and imperfect appliances until the close of the eighteenth century. The earliest lathes were of wood with dead centres, and the work was oscillated by a bow or by a spring pole. The wheel drive, giving a constant rotation to the work, appeared about the fourteenth century. On the Continent, great attention was paid to the production of mandrel lathes for ornamental turning, during the eighteenth century, and an elementary form of slide rest appeared as early as 1717 (Fig. 40). It was not, however, until 1795 that the slide rest was perfected by Henry Maudslay. Primitive lathes for cutting screws by means of a lead screw are recorded in 1569, but they were not generally used until Maudslay constructed such lathes, cutting different screws by means of change wheels, about 1800 (Fig. 41). In 1805, Maudslay made self-acting slide lathes for turning long pins.

The American invention of the turret tool holder, which greatly facilitates the production of repetition work, is attributed to Messrs. Robbins and Lawrence in 1854. This was followed by the hollow mandrel lathe, and the fully automatic lathe, with the early development of which Mr. C. M. Spencer is associated.

About the end of the eighteenth century, the drill became a machine tool by the mounting of its spindle in a frame, and applying a power drive and an automatic feed. Boring machines of the auger type were originally used for boring pipes, guns or cylinders, but the first accurate machine was introduced by John Wilkinson in 1775.

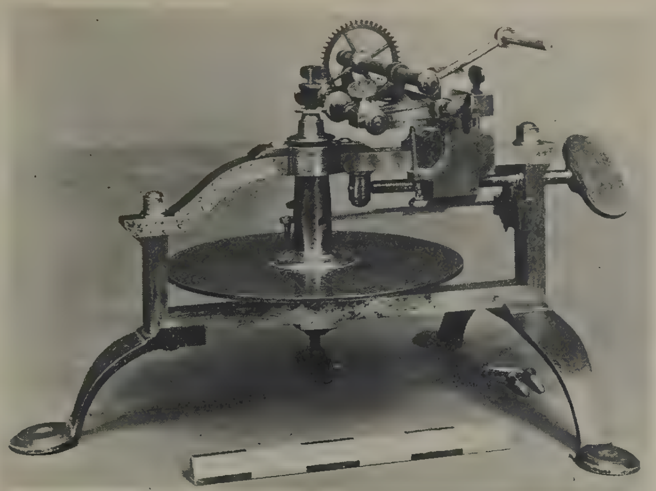

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FIG. 44. WHEEL CUTTING ENGINE, 1780

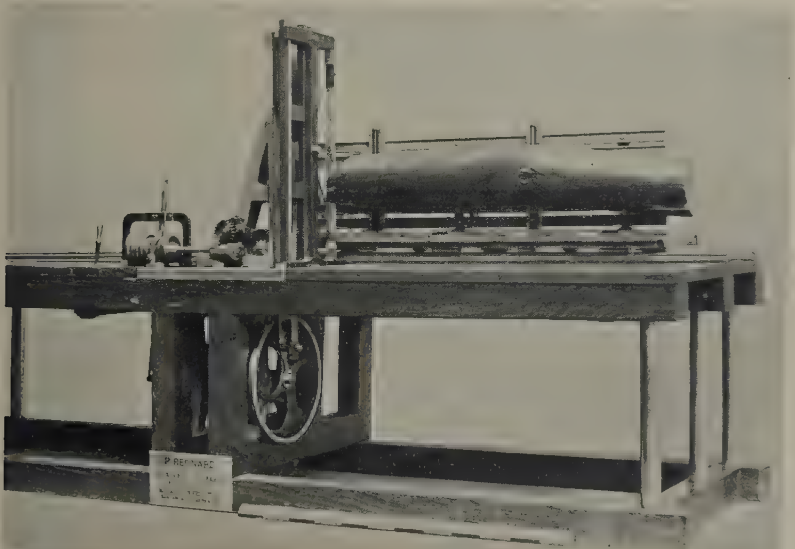

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FIG. 45. LOG SAW

Until early in the nineteenth century, flat surfaces were finished by hand, but the requirements of the steam-engine led to the invention of the planing machine, with the development of which the names of Matthew Murray, Richard Roberts (Fig. 46), Joseph Clement, and Joseph Whitworth are connected. The earliest recorded planing machine, however, was one made by Nicholas Forq in 1751.

Milling machines, in which the tool takes the form of a revolving cutter, had been used since 1670 in the form of the wheel-cutting engine, attributed to Robert Hooke (Fig. 44). Its introduction as a general machine tool, however, is credited to Eli Whitney in 1818, and its improvement to Francis Pratt in 1854. The first universal machine was made by Messrs. Brown and Sharpe in 1867.

The gear-cutting machine was developed from the early nineteenth century in both the milling and planing types, using either a former or a generating process to obtain the correct tooth form.

Although the use of grinding or abrasive processes is of primeval antiquity, yet it is only of recent years that machine grinding, by which a highly finished and accurate result is obtained, has become one of the recognized workshop operations. Grinding lathes were regularly manufactured by Brown and Sharpe from 1864, while in 1876 their first universal grinder was made. Surface and other varieties were afterwards designed by Mr. C. H. Norton, while the disc grinder was introduced by Mr. F. M. Gardner in 1890.

Wood-working machines differ from metal-cutting tools mainly in respect of their cutters, as the softer material can only be cut smoothly at low speeds by a guarded cutting edge, but may be satisfactorily worked at high speeds with unguarded cutters.

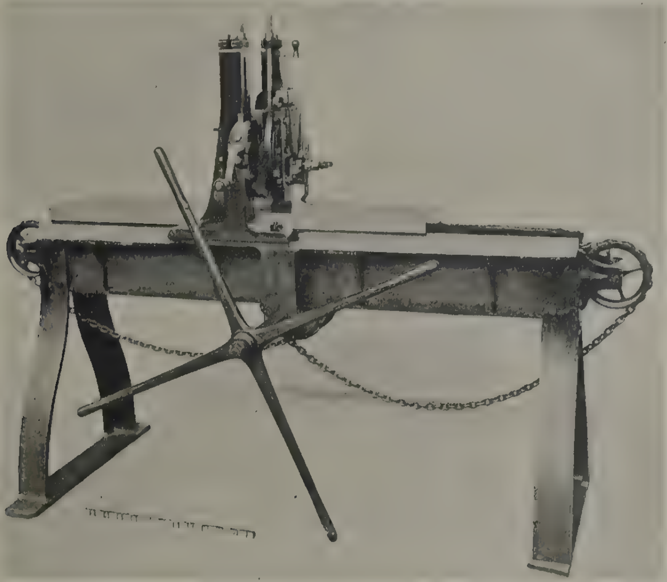

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Fig. 46. Roberts' Planing Machine, 1817

The frame saw appears to date from the fourteenth century, and is still the tool most used for converting logs into planks (Fig. 45), although the band saw, with which a higher speed is possible, is also employed for this work. The origin of the circular saw, the most expeditious of wood-cutting machines, is unknown, but it was in use about 1780, while Sir Samuel Bentham improved its construction in 1793.

The simplest form of lathe serves for ordinary wood-turning purposes, while for repetition work the copying lathe, which was introduced by Blanchard in 1818, is widely adopted.

(Conclusion)

OUTLINES OF AUTOMOBILE ENGINEERING

By J. L. Milligan, B.Sc. (Vict.), M.I.A.E.

LESSON VI

CARBURETTORS AND INDUCTION PIPES

There are many different types of carburettor. They are chiefly made by specialist manufacturers who supply them to automobile manufacturers, though some automobile engine manufacturers make their own.

These carburettors are all constructed on the same general lines. Petrol is maintained at a constant level in a float chamber and feeds through a jet into a constriction in the induction pipe. They differ very much indeed in the method by which the tendency of the simple carburettor to enrich mixture at high speeds is counteracted.

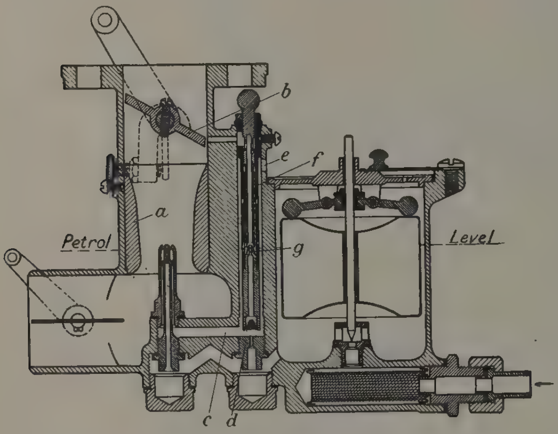


Fig. 27. "Zenith" Carburettor, Standard Type

Note. This is not a true section, but has been modified to show as clearly as possible how the carburettor functions.

Two types are here described as they are on entirely different systems. Each maker claims special features for his own apparatus, and it is an interesting fact that instruments designed on such entirely different principles give results which do not differ widely one from another.

The Zenith Carburettor ("Bavery" system). This consists of two separate carburettors with a common float chamber. The one has a choke or Venturi tube *a*, Fig. 27, below a throttle valve *b*. This has two concentric jets; the inner or main jet is fed from the float chamber and acts as a simple carburettor. The second jet is fed from a well *c* into which petrol can flow through a second jet or compensator *d*. It will be noted that the petrol in this well is open to the atmosphere *e*, so that however much the pressure in the choke is lowered the maximum amount passing is that due to the head of petrol from the float chamber level to the top of the compensator jet, and to the size of the jet. It has been found that if a simple jet in a choke tube be used, the petrol air delivery is as shown in Fig. 28, i.e. approximately a straight line passing through a point giving the minimum air speed which causes flow. If the jet is chosen so that the line is parallel to the desired petrol air ratio line, then it will be deficient at every point by the same amount of petrol. This is supplied by the compensator. There are maximum and minimum values to the air speed at which this is true, but within these wide limits this combination works.

A second carburettor comes into action when the throttle is almost closed, this consists of a pipe *f* leading into the well, mentioned above, and having a small jet hole *g* in its lower end and an adjustable air admission above the jet.

When starting, the well is full up to float chamber level, and on turning the engine, petrol is drawn up this slow running tube mixing with the air in the induction pipe and a slight leak past the throttle valve, which leak can be adjusted by a stop on the throttle lever. When the engine fires it begins to draw more petrol through this tube, the level falls slightly as it is fed through the compensator jet and the air holes are uncovered, this gives the slow running condition, the size of the jet and of the air holes being adjusted until correct. When the throttle is opened there is no longer such a high vacuum in the pipe, and the slow running setting ceases to function until the throttle is again closed.

If the air entering above the well at *e* be shut off, the compensator acts as a main jet and gives a very rich mixture at all times. The petrol in the well above the compensator gives a rich mixture on sudden opening of the throttle. In the latest forms of this carburettor there is control of air supply to the starting jet so that a very rich mixture may be obtained if desired, and also a little air can be admitted into the main jet supply in the same manner as to the compensator supply, thus reducing the volume of petrol used. This gives a mixture control from a very rich supply at small gas speed for starting cold, to the weakest possible for running down slight grades when hot.

The Smith Carburettor. This type is well known, and is designed on entirely different principles to the previous one. It consists of five separate "simple" carburettors (Fig. 29), and a sliding valve which brings them into action successively, the first, or "well" acts in a similar manner to the starting jet of the Zenith, and is used for starting and slow running, being brought into action when the throttle is closed. The other jets are in small choke tubes drilled diagonally across a cylinder *a* at such angles that the openings come at different heights. A weighted valve *b* covers these ports when the engine is at rest, or when the throttle is closed to the slow running position and no air is passing it from the main jet side. On opening the throttle, the air in the space above the sleeve is reduced in pressure and the valve rises, opening the chokes successively as the speed increases.

There are air holes *c* entering the chokes above the jets. A ring surrounding the carburettor allows these to be opened or closed, thus varying the mixture for climatic conditions.

Many other carburettors use similar schemes for starting and slow running, but differ in the method of compensation. Many systems formerly used spring-controlled valves to admit air at high speeds, but these are not now common. Others use openings in a "barrel" type of throttle to control second jets or air holes.

There is another type of carburettor which uses a small carburettor to produce a very rich mixture which is then mixed with additional air. One difficulty of to-day is the problem of obtaining a gaseous mixture with the reduced volatility of spirit and the reduced time available in the cycle of operations. This requires the addition of heat, partly to volatilize the petrol and also to supply the latent heat of volatilization. Heat is added in three ways or in combination of them. The first and oldest is to heat the air entering the carburettor by drawing it from round the exhaust pipe. This has a drawback. In hot weather the air under the bonnet has mostly passed through a hot radiator and, if it is then heated before reaching the carburettor, it may boil the petrol in the jet and cause "starving." There should always be a regulator to control the proportion of hot air. It takes time to warm the air at starting. The second method is the hot water jacket on the induction pipe. While not having the disadvantage above, it has the disadvantage of being longer in warming up but retains its heat longer, so that for moderate stops it enables working conditions to be maintained. It is clumsy and requires extra piping and joints, but is simple and satisfactory in service. The third method is the hot spot, in which a small section of the induction pipe is kept at a high temperature by either metallic contact with the exhaust pipe or by a portion of the exhaust gases being led round it in a jacket.

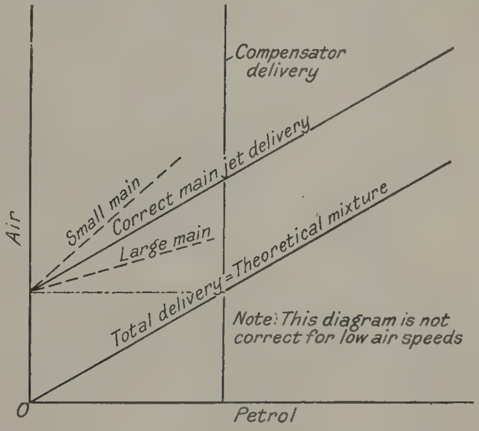


FIG. 28

If petrol has once been vaporized it may be cooled again without much condensation.

At one time wick carburettors were used, in which hot air was drawn through a mass of cotton wicks with their lower ends in petrol. This gave a gaseous mixture and was not difficult to control for mixture strength, but present-day petrol cannot be completely vaporized in this manner, there remains a small amount which accumulates and prevents starting and slow running.

INDUCTION PIPES

There are many theories about induction pipe design. Many carefully designed pipes have given poorer results than those obtained by a hastily assembled one. The main objects to be attained are—

1. Each cylinder shall have an equal volume of mixture.

2. Each cylinder shall have an equal strength of mixture.

3. There shall be little restriction to gas flow.

4. There must be no condensation in the pipe when running.

5. If condensation occurs in starting or when cold, petrol shall not drain out of the carburettor nor lodge in valve chambers or form pools.

Every possible shape has been adopted, and at present the rectangular pipe with mitred corners is often used in order to promote turbulence and prevent petrol being thrown on the walls by whirling gas (Fig. 30). At one time smooth outlines and easy bends were the correct practice. The position of the throttle valve will affect the disposition of the sprayed fuel. An arrangement as Fig. 31 is obviously bad, as the right-hand cylinders will get an advantage in slow running.

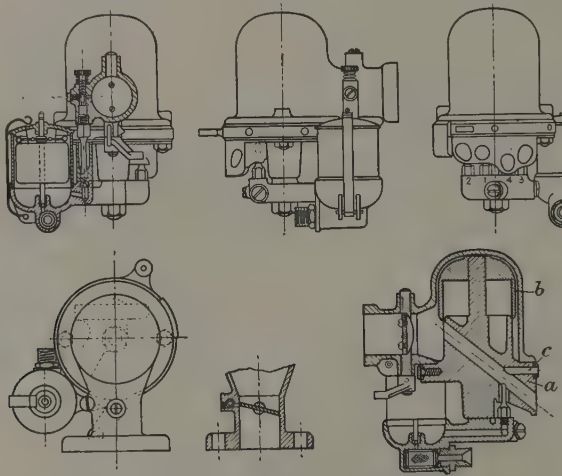


Fig. 29. "Smith" Carburettor

When petrol tanks were carried under the driver's seat on cars (they are still there on many lorries and omnibuses) the float chamber had to be kept low down in order to maintain a supply of petrol on steep hills with an almost empty tank, this gave a pipe which was good from a mixture point of view, but left the carburettor inaccessible and exposed to dirt. Petrol must be filtered between the tank and the carburettor against—

1. Foreign matter, paint, dirt off cans, and water entering by filling orifice.

2. Water condensed from the air which replaces the used up spirit; this is the source whence water mostly comes.

It is still common to put gauze filters in the filling hole, but in view of the universal use of petrol pumps the danger of foreign matter is less than formerly, and very few filters are large enough to pass petrol at the speed a pump can deliver it, hence petrol dealers usually remove the filter to fill the tank.

Tank to carburettor filters are of many types. Two common ones are chamois leather and gauze. A large number of thin copper discs separated by thin distance pieces, strung together on a rod, are also used very effectively in some proprietary filters.

Little can be said here about the practical adjustment of carburettors, known as "tuning," each make has to be tackled separately, and the carburettor makers all issue hints on the use of their own products.

LUBRICATION

Modern engines are lubricated on one of two systems. In the first or trough system, troughs are put transversely under the connecting rod bearings, and dippers on the rod end touch the oil in the troughs. The oil is pumped from the sump into the troughs to keep them full. A hole in the rod end above the dipper permits the oil to reach the big end, holes at the top side of the big end allow the oil to flow up the rod to the gudgeon pin, although the latter gets its main supply from the oil scraped by the rings from the cylinder walls. (See Fig. 18.)

The oil thrown out of the ends of the big end goes on to the cylinder walls and lubricates the piston. The main bearings are lubricated by oil pumped into pockets above them or pumped directly in by holes in the cap. Camshafts, chains, etc., are lubricated by splash from the big ends or by separate leads from the oil pump. This system uses very low pressures, not over 2 lb. per sq. in. It is very efficient and simple. Troughs should be adjustable and curved so that the amount of " dip " is not altered by running on the side of a heavily cambered road. The dip varies on different makes from $\frac{1}{32}$ to $\frac{1}{8}$ in. Trough systems should have large bore pipes, and the feed to the troughs should not blow the oil away from the dippers when running quickly.

The second system is the complete forced system by which oil is pumped into each main bearing and then by holes drilled in the crankshaft to the big ends. Sometimes a pipe is led up the rod to the gudgeon pin, but this is not necessary. Camshaft bearings, cams, chains, are also fed in this system. A very efficient filter is necessary. This can be on either suction or pressure side of the pump. It is usual now to fit two filters, a coarse one on the suction side and a finer one on the pressure side. The latter should be readily removable, without much difficulty, and the suction one should be removable when flushing out the engine. Pressures in this system are about 20 lb. sq. in. Recently a type of filter in which a small percentage of oil is by-passed through a special fine filter has become popular, and is likely to be more general in its application.

Oil pumps are almost exclusively of the gear wheel type. They should be buried in oil and, in order to run silently, should be cut with a lot of top and bottom clearance and backlash, but should be a good running fit in the casings.

Oil piping should preferably be of steel with welded joints. No general law can be laid down as to suitable lubricants, it can only be settled by trial, but in general an oil should be used which will not gum up in cold weather yet not burn away in heat. Some cars require much thinner oil than others.

WATER CIRCULATION

This is done either by natural circulation or by a pump. The former is simpler but adds weight, as large pipes are necessary and also more water. It will not work if the system is not full of water, so that a leak soon causes boiling. There must be a certain head of water in the top tank to take the water through the tubes. The lower tank of the radiator should be reasonably deep, or else one side of the radiator will do all the circulation. The distribution of cold water to the cylinders and the collection of hot water from them requires to be thought out carefully to give a good flow under the valves, especially the exhaust valve, and at the same time give uniform circulation to each cylinder with no short cut from pump to top tank. Pumps are almost without exception of the centrifugal type, the only exceptions are those of the screw type combined with a fan drive.

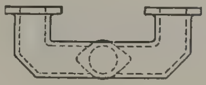


FIG. 30

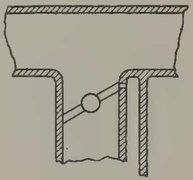


FIG. 31

Cooling fans are mostly belt driven as the shocks caused by accelerating and retarding engines put a severe load on the driving gears. The fan is seldom necessary on a car in this country for six months of the year. Radiator thermometers are now in very general use. Thermostats are also fairly common, these consist of a thin corrugated bellows of copper containing a little air with some spirit and sealed. This is included in the top water pipe. A valve fixed to the bellows closes the outlet from pipe to radiator. The other end of the bellows is fixed to the pipe itself. When the temperature of the water reaches the boiling point of the spirit the vapour expands the bellows and opens the valve. There is thus automatic control of water jacket outlet temperature, this permits of rapid " warming up " in starting off.

The delivery capacity of pumps varies very greatly in different makes, in general the more rapidly water is circulated the less chance of stagnant spots. On the other hand, a high speed may permit eddies in corners. Pumps are often driven in tandem with magnetos to damp out the shocks on the drive of the magneto.

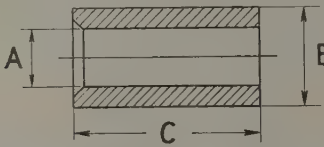
JIGS AND TOOLS

By LEO KELSEY

LESSON IX

STANDARD JIG AND FIXTURE COMPONENTS

Steels. As the cost, in most cases, of jigs and fixtures of all types is largely made up of the time spent in making them, the materials represent but a small proportion of the total cost of such articles. It is, therefore, more economical to use the best materials in the beginning, as it frequently happens that it is not possible to discover whether a piece of material is sound throughout its section until a considerable sum in labour has been spent upon it.



A	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$
B	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	1	$1\frac{3}{16}$	$1\frac{3}{8}$	$1\frac{5}{8}$	$1\frac{7}{8}$
C	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{5}{8}$

FIG. 83

Regarding castings for jigs and fixtures, many foundries now make a special class of iron for tool work, which is close in texture. Special precautions are taken to eliminate blow holes. Should it happen that the patterns are sent to a foundry which has not done this class of work before, the best soft grey cylinder iron should be specified. Only the best quality soft mild steel should be used, and it is more economical to use bright drawn than black, as in many cases parts of jigs and fixtures can be constructed of this material without any further finishing. Steel which has to be hardened should be of the very best quality obtainable, and the makers' instructions on heat treatment rigidly adhered to. Whenever possible, precision ground stock, as manufactured by the more reputable makers, should be used.

Bushes. A table of standard drill bushes appears in Fig. 83. Where possible, the latter—or a similar table—should be followed, although in many instances it may not be possible. It may be noted here that many designers prefer heads on their drill bushes as illustrated at Fig. 84, but individual requirements should decide whether this is necessary or not. Should slip bushes be used, however, heads are necessary.

Dowels. Dowels should always be made of even sized silver steel rod, i.e. $\frac{1}{16}$ in., $\frac{3}{32}$ in., $\frac{1}{8}$ in., $\frac{5}{32}$ in., $\frac{3}{16}$ in., $\frac{1}{4}$ in., $\frac{5}{16}$ in., $\frac{3}{8}$ in., $\frac{7}{16}$ in., $\frac{1}{2}$ in. The holes in which the dowels fit should be reamed to a smooth finish, and the dowels must be a good driving fit therein.

Dowels should never be larger than the screw which is holding together the two surfaces which the dowels are locating. To make them larger merely weakens the part they are in, without increasing the resistance to movement, which largely depends on their fit and not their diameter.

Screw Threads. All screw threads in jig and tool work should be standardized. Unless the metric system is used in the factory, screw threads should be standardized on the following lines—

For holes less than $\frac{1}{4}$ in. in diameter use
British Association Nos. 0, 2, 4, 6, 8.
For holes over $\frac{1}{4}$ in. and less than 1 in. in diameter use
Whitworth.
For holes over 1 in.
British Screw Fine (B.S.F.).

Square threads for machine slides, etc., should either be of the standard square thread form or standard American acme thread form. The latter thread is the easier of the two to cut, and should be used unless the thrust on the screw is very considerable, when the square thread is preferable.

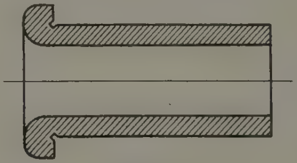


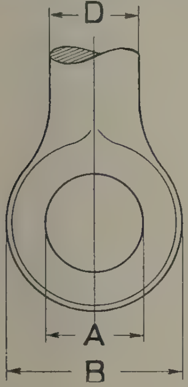
FIG. 84

Nuts, Wing Nuts, Bolts, and Eye-bolts. In the majority of cases, standard nuts and bolts can be more satisfactorily bought from firms specializing in their manufacture. These, however, must be of first-class quality, up to size on the flats and of full thread.

In Fig. 85 is shown a table of standard eyebolts which has been found satisfactory in practice. The diameter marked *B* is turned as a sphere and the flats are milled afterwards.

Tenons. A table of standard tenons is given in Fig. 86. Dimension *A* should be made to suit individual machines on which the fixtures are to be used.

Preparation and Dimensioning of Tool Drawings. All tool drawings should be on the standard quarter, half and full imperial, or double elephant. After the general arrangement of the tool has been drawn, with the component in position (the latter usually drawn in colour), the important assembled dimensions of the tools should be put on, after which each part of the tool should be made the subject of a separate, fully dimensioned drawing. On the general arrangement drawing, a list should be compiled of every component of the tool and its proper name. Reference should be made by means of a number, which should apply to one component only.

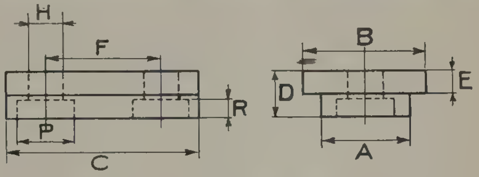


D	A	B
½	9/16	1
⅝	11/16	1¼
¾	13/16	1½
⅞	31/32	1¾
1	1 3/32	2
1⅛	1¼	2¼
1¼	1⅜	2½

FIG. 85

Checking of Design and Tool Drawings. The design of every form of tool should be criticized systematically. In many cases it is not possible to criticize a tool which may be one of a number used on one component, without reference to the preceding and succeeding tools. Therefore, the first point to criticize in any design is, whether the tool in question is the correct one to use with reference to the layout of the component: the second point is to see that the locating surfaces are correct, and in accordance with the tooling scheme of the component. The third point is to observe whether the tool will perform the function desired of it in the most economical and reliable manner. The fourth point is to see that the component can be easily inserted and removed from the tool. The fifth point is that the clamps will secure the component correctly in relation to the locating surfaces, and also that they will allow the preceding condition to be satisfied. The sixth point is to be sure that as much clearance as is practicable is allowed for possible variation in the size of the component.

If the above conditions are satisfied, there should only be the individual peculiarities of the design to contend with from a synthetically critical point of view.



A	B	C	D	E	F	H	P	R
⅜	⅝	¾	5/16	⅛	ONE IN CENTRE ONLY	3/16	21/64	9/64
½	¾	1	5/16	⅛	½	3/16	21/64	9/64
⅝	1	1¼	⅜	3/16	¾	"B" DRILL	27/64	3/16
¾	1 3/16	1½	7/16	¼	1	"B" DRILL	27/64	3/16
⅞	1 5/16	1½	7/16	¼	1	"F" DRILL	25/64	7/32
1	1 7/16	1⅝	½	¼	1 1/16	"F" DRILL	25/64	7/32

FIG. 86

Milling jigs should always be heavy, and if the cut to be taken on the component is at all heavy, the fixture must be massively designed.

Most tool drawing offices have their own peculiarities in the production of drawings, but the following remarks will guide the student in this respect—

Keep dimensions outside the various views as much as possible, and see that the dimension lines are carried to the part in question.

When dimensions are figured, they should be done so as to give the shop the maximum amount of information. Never give dimensions of precision in fractions, but always in decimals.

Put tool detail numbers in large circles.

Except in the very simplest tools, always make a separate drawing for each detail.

Dimensions must all read in the same direction, so that the drawing can be read without being turned round or upside down.

THE THEORY OF HEAT ENGINES

By E. B. Cole, B.Sc.

LESSON XII

THE TOTAL HEAT-ENTROPY CHART—(*contd.*)

The construction of the complete chart was described in the last lesson, but for steam, in actual practice, only a portion of the diagram as indicated in Fig. 34 is required. For refrigeration work with CO_2, the portion of the chart for this substance in the neighbourhood of the critical point is most important.

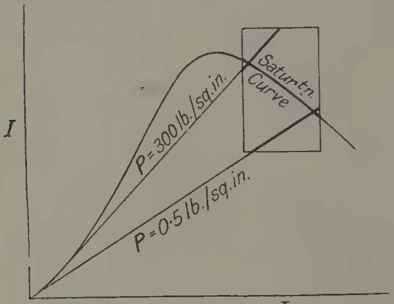


Fig. 34

Let us now examine the use of such a diagram.

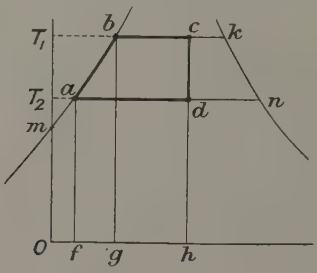


Fig. 35

We have seen that the work done on the Rankine cycle, or the "Rankine Heat Drop," is given by $(H_1 - H_2)$ between any two levels (1) and (2), where H_1 = the total heat of the steam before the adiabatic expansion, and H_2 = that after expansion.

The T–ϕ diagram for this cycle, with steam initially wet, is reproduced in Fig. 35, and the corresponding diagram on the Mollier chart is shown in Fig. 36, where—

I_a = total heat of 1 lb. of feed water at lower temperature T_2, which is ≒ (nearly equal) to area *omaf* on $T - \phi$ diagram.

(I_a is actually equal to this area + the work done by feed pump per lb., but we have agreed to neglect this slight difference between I and H.)

"*ab*" represents the warming of the water up to the higher temperature T_1, the pressure being raised at the same time to P_1. Thus,

I_b = total energy of 1 lb. of water at temperature T_1,
≒ area *ombg*.

"*bc*" represents evaporation at this constant temperature and corresponding pressure to a dryness given by *bc*/*bk* on either diagram.

I_c = total energy of 1 lb. of steam of this dryness at the higher temperature T_1,
≒ area *ombch*.

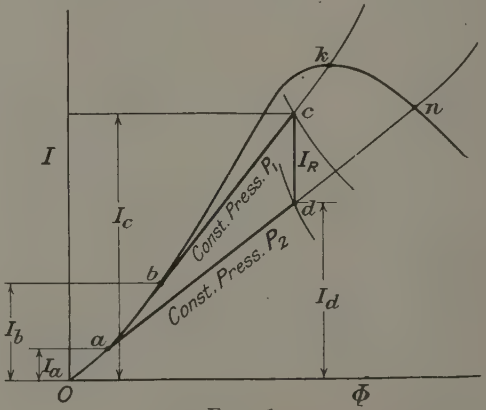


Fig. 36

Adiabatic expansion is shown by the vertical line *cd* (constant ϕ) on either diagram, and

I_d = total energy of 1 lb. of steam, of the quality x_d attained after the expansion, at the lower temperature level T_2,
≒ area *omadh*.

Then from the $T-\phi$ chart, the work done on the Rankine cycle = area *abcd*,

$$= H_c - H_d,$$

which is approximately equal to $I_c - I_d$, which on the $I-\phi$ chart is given by I_R, the length of the vertical line *cd*.

Hence, on the $I-\phi$ diagram, the Rankine heat drop is obtained by *direct measurement of a vertical straight line*, drawn from the point *c*, representing the initial quality of the steam at the higher pressure, down to the constant pressure curve for the lower pressure. The final dryness is also directly indicated.

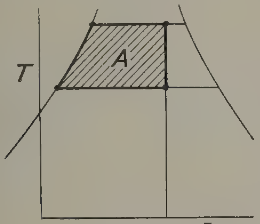


FIG. 37

We shall proceed to enlarge upon and use these results in the following sections—

FLOW OF STEAM THROUGH NOZZLES

In discussing the throttling of steam as represented on the $T-\phi$ chart, we showed that the heat energy, represented by the shaded area *A*, Fig. 37, is converted into kinetic energy of the steam jet, which, due to internal friction and eddies, is reconverted into heat energy at the lower pressure.

This shaded area the student will recognize as equivalent to the work done by an engine operating on the Rankine cycle. If, however, in allowing the steam to expand from one pressure to a lower, we control the steam jet by means of a properly shaped nozzle, we can make use of the kinetic energy of the jet so formed instead of wasting it in eddy motion. The steam will then have the same dryness after the expansion as it would have after adiabatic expansion on the Rankine cycle, the jet energy can be utilized, so that, as previously stated, we may consider such expansion in a nozzle as *adiabatic* and at constant entropy.

Thus, the increase in kinetic energy of the steam above that possessed initially—or, if this latter be negligible—the kinetic energy of the jet will be given by

$$\underset{\text{(heat units)}}{K.E. \text{ per lb.}} = \frac{v^2}{2gJ} = H_1 - H_2$$

or the velocity of jet $v = \sqrt{2gJ\ (H_1 - H_2)}$.

Let us examine this in another manner by considering the energy possessed by 1 lb. of steam flowing with a steady velocity V ft. per sec.

The energy possessed per lb. will consist of—

1. The kinetic energy $= \dfrac{v^2}{2gJ}$ heat units.

2. The pressure energy and internal energy, the sum of which we have previously called the "total energy" I, i.e. $I = E + \dfrac{PV}{J}$.

Now, since heat and work are mutually convertible and the whole energy possessed by a substance remains constant, we may therefore write

$$\frac{v_1^2}{2gJ} + I_1 = \frac{v_2^2}{2gJ} + I_2$$

which represents a change in the substance from an initial state (1) to a final (2). Thus—

$$\frac{v_2^2 - v_1^2}{2gJ} = I_1 - I_2$$

or, if the fluid be initially at rest,

$$\frac{v_2^2}{2gJ} = I_1 - I_2$$

i.e. the velocity of the jet

$$v = \sqrt{2gJ\ (I_1 - I_2)}.$$

If we assume that the expansion is adiabatic then $I_1 - I_2$, which is approximately equal to $H_1 - H_2$, represents the work done on the Rankine cycle per lb. of steam, and for our purpose here the steam will be considered to flow *adiabatically and without friction*. Hence, the velocity of the jet so formed by the steam expanding through the nozzle can be obtained for any given pressure drop directly from the $I-\phi$ chart, or approximately by calculation of the Rankine heat drop.

Let us determine the velocity attained by steam initially dry and saturated on expanding adiabatically from a pressure of 150 lb.-sq. in. absolute to various back pressures.

The results obtained from measurement on the $\phi - I$ chart are tabulated below, but the student is advised to calculate the velocity from determination of the Rankine heat drop, so that he may make himself thoroughly familiar with both methods—

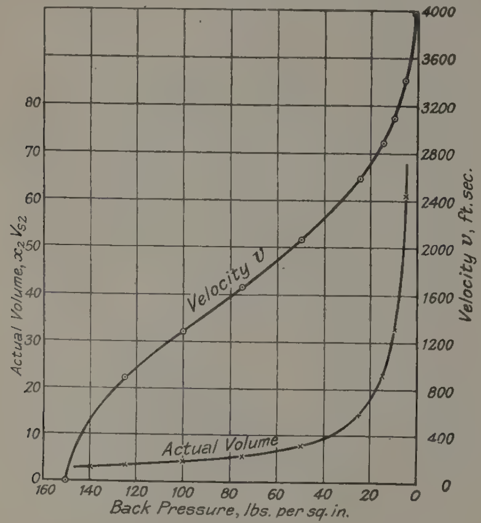


FIG. 38

Back Pressure Lb.-sq. in.	Final Dryness x_2	$I_1 - I_2$ B.Th.U.	v ft.-sec.
125	·986	15·8	890
100	·971	32	1,276
75	·952	55	1,660
50	·929	85	2,060
25	·894	134	2,586
15	·871	167	2,890
10	·856	192	3,100
5	·830	233	3,415
1	·780	316	3,980

In the determination from the $\phi - I$ chart, a vertical line is drawn from the point on the "saturation line," cut by the constant pressure curve for 150 lb.-sq. in., and the intersections of the various back pressure curves give the corresponding dryness fractions x_2, while the lengths of the line give $I_1 - I_2$. A scale graduated direct in corresponding velocities is often printed by the side of the total heat scale.

The graph, Fig. 38, shows these results, the velocity attained being plotted against the back pressure.

To Find the Cross-sectional Area of the Nozzle at any point of known back pressure.

For any given final pressure we have seen how we can determine the dryness fraction x_2. Thus, if w lb. of steam are flowing per second, and V_{s2} is the specific volume of the steam at the given pressure, then the total volume passing this section per second will be $w \cdot x_2 V_{s2}$.

This volume will be passing with a velocity

$$v = \sqrt{2gJ(I_1 - I_2)} \text{ ft.-sec.}$$

Hence, if A = cross-sectional area in sq. ft.

$$\text{Then} \quad A = \frac{wx_2V_{s2}}{v}$$

The following table gives the values of A necessary, as expansion proceeds, to pass 1 lb. of steam at any section at which the back pressure is as indicated, the initial conditions being as before—dry saturated steam at 150 lb.-sq. in. absolute—

Back Pressure at Section	Specific Volume V_{s2}	$A = \frac{x_2 V_{s2}}{v}$ sq. ft.	Discharge in lb. per sq. ft. per sec.
125	3·583	·00397	252
100	4·429	·00337	297
75	5·81	·00333	300
50	8·51	·00384	260
25	16·30	·00563	178
15	26·27	·00792	126
10	38·38	·01058	94·5
5	73·33	·01785	56·0
1	333	·0652	15·4

The areas are those required per lb. of steam, and the reciprocal of this shown in the fourth column will give the amount of steam that could pass per second per square foot of area.

These results are plotted against the back pressure in Fig. 39, and the student is asked to examine the graphs carefully.

On Fig. 38 also, the actual volume ($x_2 V_{s2}$) at any section of given back pressure is plotted.

Now, while we are enabled by this method to find the area of cross-section for any back pressure, we have no means, as yet, of determining the actual *shape* of the nozzle, since we do not know how the pressure varies along the axis of the nozzle. If we assume a uniform pressure drop along the axis, then the nozzle form will be that shown in Fig. 39, where the diameter has been plotted against the uniform back pressure scale. However, for our purpose here, the actual shape does not matter.

The area of cross-section depends on the ratio xV_s/v, and, examining Fig. 38 for these two quantities, we see that as the back pressure is reduced the velocity v increases and, therefore, the area required tends to diminish, but at the same time the volume is increasing which will require an increase in area.

At first, however, the rate of increase in velocity is greater than the rate of increase in volume, which means that the area gradually diminishes, but after a certain point, here about a back pressure of 87 lb.-sq. in., the rate of increase in volume exceeds that of the velocity, and hence the area required will begin to increase. Thus, the nozzle first *converges* to a *throat* and then *diverges*.

Consider now the discharge curve for a back pressure above that at which the throat occurs, say, 125 lb.-sq. in., the nozzle required would be the portion AB, and if the exit area at B were 1 sq. ft., then 252 lb. steam would be discharged per second. The shape of the nozzle would be convergent. Similarly, for any other back pressure down to that at which the throat occurs the nozzle is convergent, and if the exit areas were in each case made equal, we see that the quantity of steam which can be passed increases as the back pressure diminishes. But after the throat section is passed the nozzle diverges, and the quantity of steam per sq. ft. per second now diminishes.

Thus, given any particular nozzle, the mass flow per second can be increased by reducing the back pressure to a certain definite value, at which the throat or minimum area is formed. Reducing the back pressure below this value does not increase the mass flow, this being determined by the smallest section, but does increase the kinetic energy.

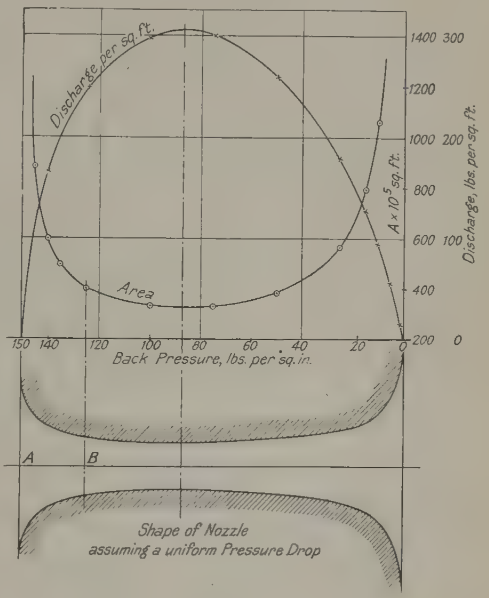


FIG. 39

In our case, the value of this *critical* back-pressure is 87 lb.-sq. in., or about 0·58 times the initial pressure (150 lb.).

STEAM TURBINES

By W. J. Kearton, M.Eng., A.M.I.Mech.E., A.M.I.N.A.

LESSON VIII

CONSTRUCTIONAL DETAILS OF IMPULSE STEAM TURBINES—(*contd.*)

Blades. It was natural that in the early turbines the blades should be formed by bending sheet metal into a curve, such that the tangent to the curve at the inlet edge of the blade would coincide with the direction of inflow of the steam relative to the blade, Fig. 31. The sudden change of direction suffered by the steam when passing through the curved channel between the individual blades causes the steam to be compressed, and the width of the flowing stream would tend to diminish somewhat, as indicated roughly in the left-hand side of Fig. 31.

The idle space immediately in front of the blade would be full of eddies, and the energy absorbed in sustaining these eddies would be a dead loss. Usually, the width of the channel z is made more or less constant, thus giving a blade section somewhat like that shown in the right-hand side of Fig. 31. With equal inlet and outlet angles, the radius $r = R - z$.

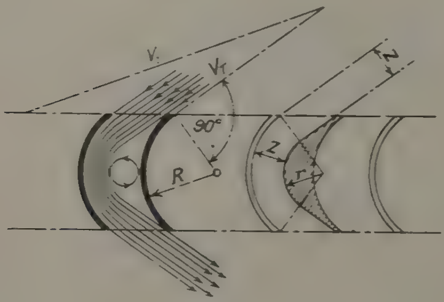


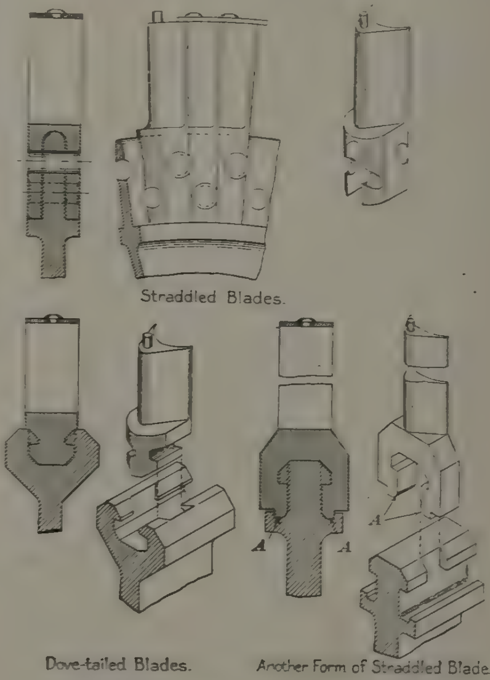
Fig. 31

Attachment of Blades to Rotor. Owing to the high peripheral velocity common in modern steam turbines, the centrifugal force tending to wrench the blade from the rotor is fairly considerable, especially with long blades. The attachment of the blades to the rotor is thus a most important detail.

The principal desiderata may be enumerated as follows—

1. The method of attachment and the dimensions of the parts should be such that the factor of safety in the principal parts is approximately constant.

2. As far as possible, the machining and other operations should be free from "the personal equation." The amount of fitting should be a minimum.



Straddled Blades.

Dove-tailed Blades.

Another Form of Straddled Blade

Fig. 32. Turbine Blade Attachments

3. It is an advantage to be able to renew any individual blade without disturbing the others.

Three common forms of attachment are illustrated in Fig. 32.

In the "straddled" blade, the blade root is forked and forced on to a projecting lip which is,

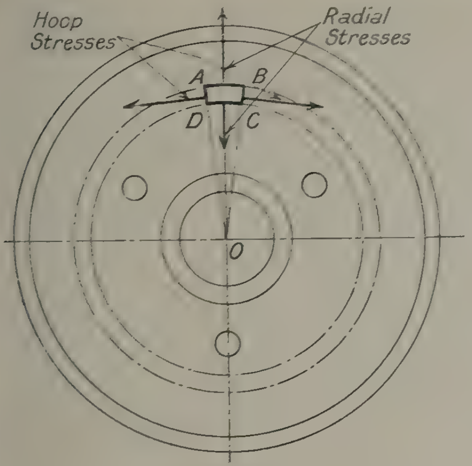


FIG. 33

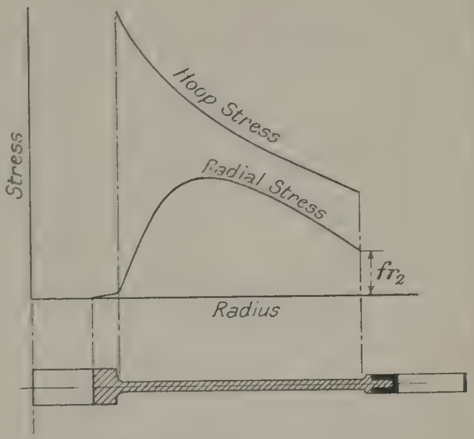


FIG. 34. STRESS DISTRIBUTION IN ROTOR DISC OF CONSTANT THICKNESS

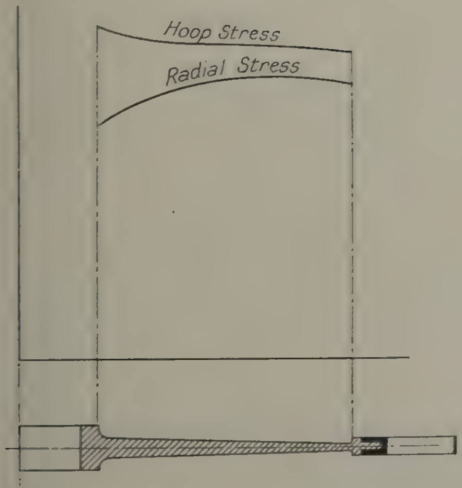


FIG. 35. STRESS DISTRIBUTION IN ROTOR DISC OF VARIABLE THICKNESS

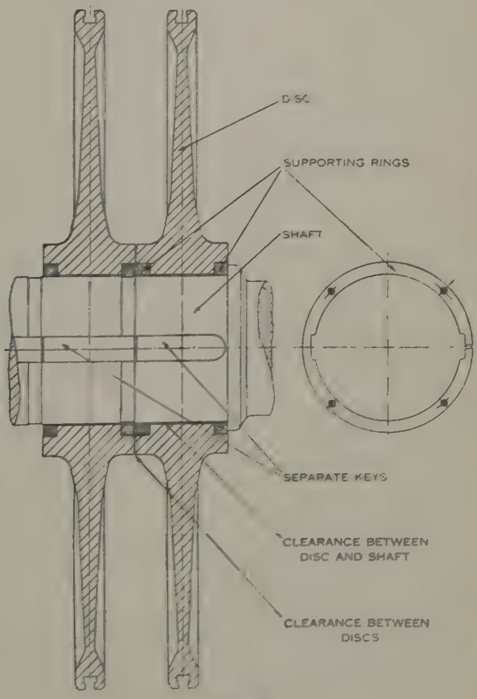


FIG. 36

practically speaking, a continuation of the disc. Each blade is secured by the equivalent of one rivet in double shear. The rivet holes are slightly countersunk, and the actual filling of the countersink is done by a special machine. The blades are attached to the rotor in a very efficient and direct manner, and any one blade may be removed and renewed without disturbing any of the others.

The dovetail method is used almost exclusively for velocity-compounded wheels, but there are still many builders who use it for the single-row wheels. It does not give such a simple state of stress as the straddle method. In order to insert the blades during assembly, it is necessary to cut a slot or port in the rim at some point. The blades are then driven round to their correct positions until the whole rotor is bladed. The "port" is then filled up by a special filling piece and, of course, there is a gap in the blade ring at this point. If it is desired to remove a blade that happens to be on the side opposite the slot, then it is necessary to remove half the blades. Another disadvantage of this type of fastening is the relatively wide rim that is required.

The modified straddle attachment shown in Fig. 32 is adopted for long blades and high peripheral velocities. The periphery of the disc is turned to a T-section, and at one point slots are cut in the flange so that the blades may be slipped on. Projections *A* on the blade root fit into recesses in the rim and prevent the fork opening out.

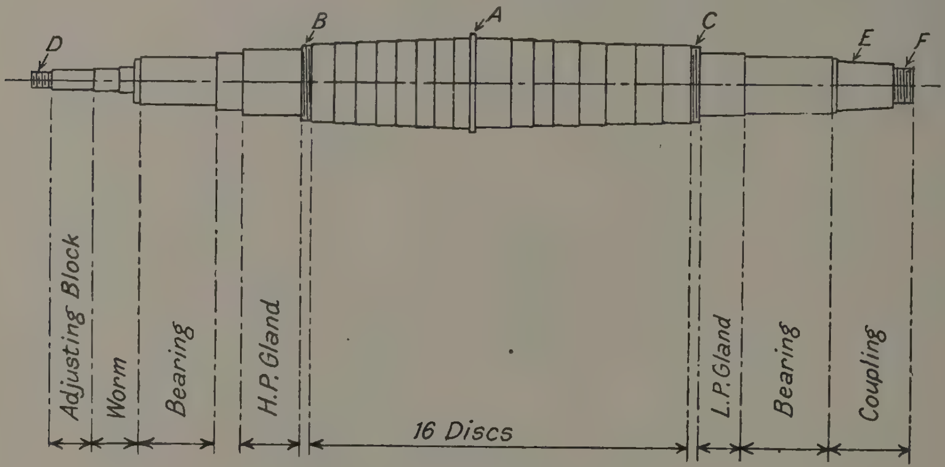


FIG. 37. H.P. TURBINE SPINDLE

Rotors. The rotor disc serves to transmit the torque from the blades to the shaft, and also to carry the blades and hold them against the forces due to inertia. It is the latter function of the rotor disc and the necessity of resisting the inertia forces due to its own mass, which influences its shape.

The only stresses of any importance in a rotating disc are the *hoop stresses* and *radial stresses*. The meaning of these terms will be understood by referring to Fig. 33. Imagine an "element" of the disc, *ABCD*, bounded by the circular arcs *AB* and *CD* and the radial lines *AD* and *BC*. The stresses perpendicular to *AB* and *CD* are termed radial stresses, and those perpendicular to *AD* and *BC* are called tangential or hoop stresses.

The variation in the magnitude of these stresses depends, as one would expect, on the cross-section of the disc. In a rotor of which the disc proper is of constant thickness, as shown in Fig. 34, the radial stress near the hub will be very small, and near the rim will have some value f_{r2} owing to the centrifugal force on the blades. The distribution of hoop stress is shown by the curve, and it will be seen that the greatest value of this stress occurs in the metal near the hub of the rotor.

If the most economical use is to be made of the material of the rotor, then the stresses should be fairly uniform throughout the section. This ideal is more or less realized in a well-designed disc by varying the thickness as shown in Fig. 35.

In order to preserve equality of steam pressure on both sides of the discs, it is usual to bore pressure-balancing holes in the discs as shown in Fig. 33. Such holes weaken the disc in their neighbourhood.

The wheels are sometimes forced on to their seats on the shaft by means of hydraulic presses. Some amount of force fit is necessary ; otherwise the bores of the wheels would open out to such an extent that they would become loose on the spindle. Some makers, on the other hand, interpose a hard bronze tapered bush between the wheel bore and the shaft, and claim that by using dissimilar metals, " biting " or " seizing " is prevented.

Another method, which is utilized by the English Electric Co., is illustrated in Fig. 36. The hub of the wheel is bored out slightly larger than the diameter of the shaft. Recesses are bored in each end of the hub for " supporting rings," each of which is gauged to suit the diameter of the shaft and recess. It is stated that when the discs are rotating at their normal speed, the strains in the hub are such that the ends of the hub nip the supporting rings so that the discs are held quite tight. The wheels are located axially by every second ring which fits into a recess in the shaft. The torque is transmitted from the central portion of the hub only, through two sunk keys. As there is a radial clearance between the shaft and the bore of the wheel, and also an axial clearance between each disc and its neighbours, shaft flexure does not tend to cause slackness.

Spindle. The spindle is a high-grade steel forging machined all over, and sometimes bored from end to end for the purpose of inspecting the centre of the forging.

The size, i.e. the diameter of the spindle, is not decided according to the magnitude of the shear stress produced by torque, but so that the normal running speed will be sufficiently far removed from the critical speed. This condition usually involves a spindle of very liberal dimensions, as will be apparent from Fig. 37.

When the discs are forced on the spindle by hydraulic jacks or a press, the diameters in the central portion of the spindle are " stepped " from each end to the central collar *A*, the discs being secured finally by fine-threaded nuts screwed on *B* and *C*. The spindle is screwed at *D* for the nut securing the adjusting block collar and also for the emergency governor, and it is tapered at *E* for the solid coupling which is secured by a nut screwed on *F*.

The complete rotor should be balanced statically or dynamically.

STEAM BOILERS AND THEIR ACCESSORIES

By Percy E. Rycroft, M.B.E., M.I.E.E.

LESSON VIII

MECHANICAL STOKERS—(*contd.*)

The self-contained travelling grate stoker manufactured by the Underfeed Stoker Co., Ltd., is of particular interest from the fact that a complete forced draught equipment including both fans and an electric motor, together with stoker driving gear, form an integral part of the apparatus.

In principle the stoker is similar to a chain grate so far as carrying the fuel through the furnace is concerned, but the details of construction have many points of difference.

The grate surface is composed of fingers or inset pieces carried by frames in which they are uniformly spaced, and provide a means for admitting a suitable supply of air through the fuel bed.

The grate sections are supported by angle shaped carrier bars which extend across the full width of the grate and are mounted on endless chains arranged at either side.

These, in turn, gear with suitable driving sprockets operated by the driving mechanism and having a range of eight speeds.

The box-shaped sections, when travelling through the furnace, form rectangular ducts through which air is forced under pressure from the main fan ducts at either side.

Suitable variation in the amount of air pressure required along the length of the grate is obtained by means of longitudinal triangular dampers, which can be placed in any desired position so that the air can be admitted progressively from front to rear of the stoker.

The fans—one for either side of the grate—together with the electric motor for driving them, are mounted on a common shaft situated immediately in front of the stoker hopper and, as already indicated, power for operating the stoker drive is also obtained from the same source.

The general arrangement of the complete unit will be apparent from Fig. 49, and the apparatus is suitable for burning a wide range of fuel, including low volatile coal and anthracite, coke breeze and low grade bituminous slacks, containing up to 50 per cent of ash, as well as the higher quality of fuels.

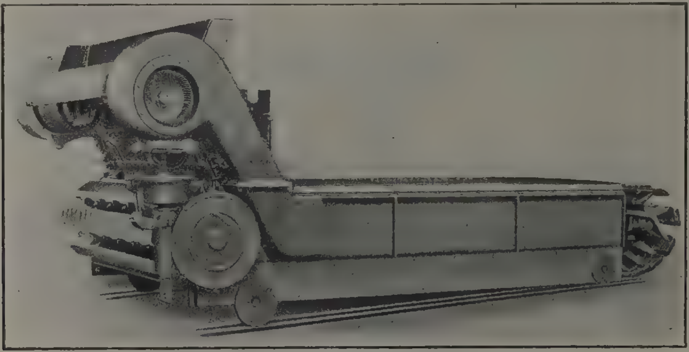


FIG. 49. SELF-CONTAINED TRAVELLING GRATE STOKER
(Underfeed Stoker Co., Ltd.)

Bennis Chain Grate. Another stoker of the chain grate type is made by Messrs. Edward Bennis & Co., Ltd.

In essentials it differs little from preceding examples, the grate proper taking the form of an endless chain built up from a number of relatively small cast-iron links connected together transversely by steel rods, and the whole being driven by suitable mechanism mounted at the front of the stoker.

The stoker links are of a special patented design rounded and "halved" together at either end. By this arrangement the grate surface remains unbroken even when the links are passing round the sprockets at either end of the grate. The sides of the links are corrugated and special attention is paid to accurate air spacing.

Another particular feature of this link is the provision of a dowel on either side of it, each being towards its own end and loosely fitting into a corresponding recess in the adjacent link.

This arrangement prevents a link falling out of position should a breakage occur at any time, and is a useful precaution against complete jamming of the grate due to this cause.

The grate is ordinarily designed to operate under induced draught only, but can be adapted for forced draught conditions when this is required.

The Riley Retort Stoker. The general principle of the retort type of stoker have already been referred to, and the Riley grate consists of a series of channels or retorts extending from the front of the boiler to the ash plates at the rear and with an inclination to the horizontal of about 25°.

The sides of the retorts are designed for a reciprocating movement and carry a number of "stepped" cast-iron blocks which form a continuous grate surface between the retorts, but are cast with suitable openings so that air can be forced under pressure into the fuel bed.

The retorts are spaced about 18 in. apart, and the fuel is forced into the upper end of the channel by means of a mechanically operated ram or plunger.

The amount of coal delivered at each movement of the ram is regulated by a stroke adjuster, and the travel of the reciprocating sides can also be varied as required.

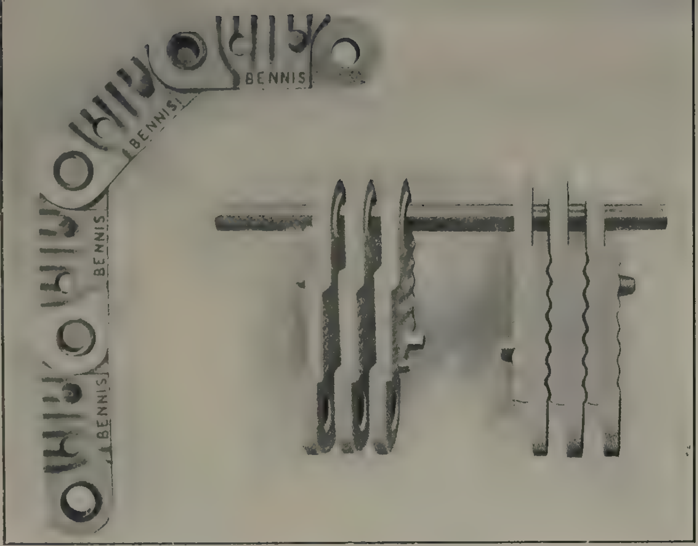


Fig. 51. Bennis Chain Grate Stoker Links

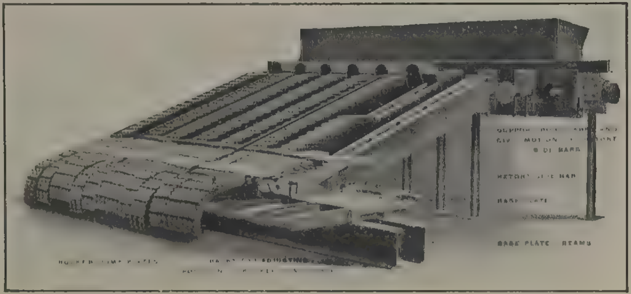


Fig. 52. Riley Self-dumping Underfeed Stoker

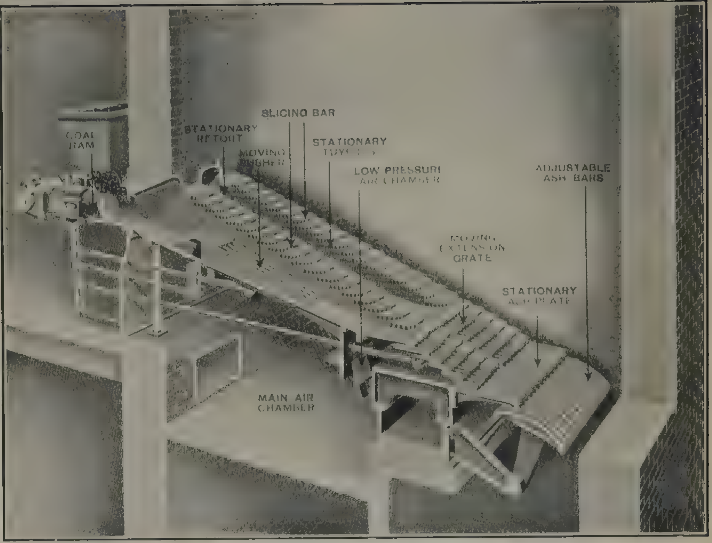


Fig. 53. Erith-Roe Stoker

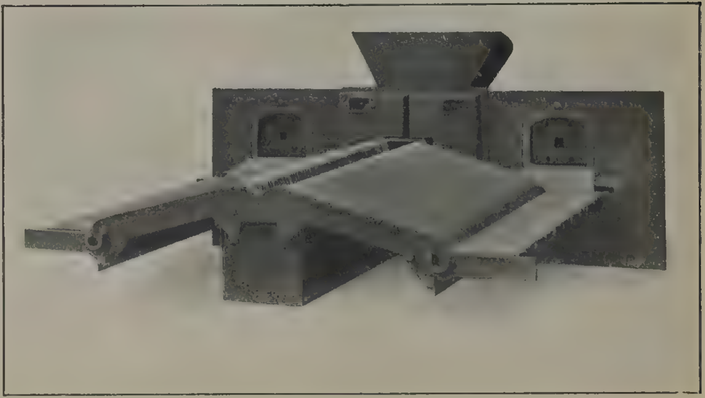


Fig. 54. Interior View of Class "E" Stoker
(Underfeed Stoker Co., Ltd.)

In action the combined mechanism, assisted by gravitation, results in a joggling motion, which carries the fuel slowly through the furnace and on to the ash plates which extend across the entire width of the stoker at the rear.

Here there are "rocker" plates mechanically operated which, having a combined horizontal and vertical motion, continuously crush the residue of ash and clinker against a bridge wall, the space between which and the rocker bars being adjusted by means of an exterior hand-wheel.

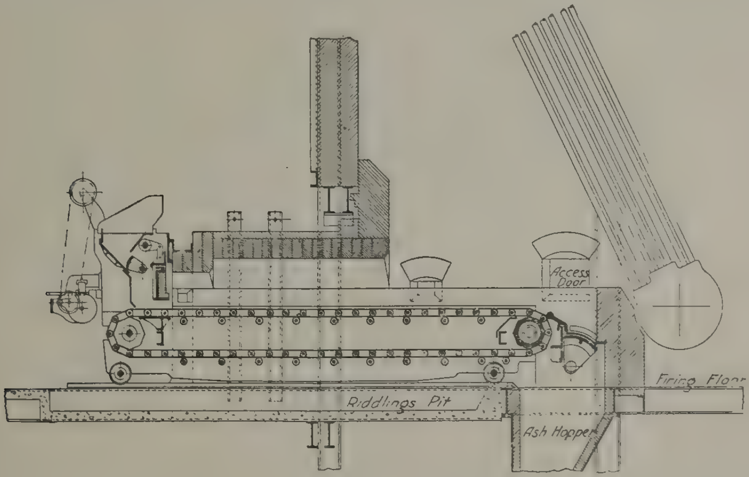


FIG. 50. BENNIS CHAIN GRATE STOKER
(Longitudinal section)

Erith-Roe Stoker. In the Erith-Roe apparatus the general principle of the retort type of stoker is retained, but a number of special features have been introduced.

The most notable modification is the introduction of several additional movements to the grate parts, all independently controlled and all capable of being individually adjusted.

Special "pushers" are fitted in the bottom of each retort to ensure a positive movement of the fuel, and to assist in breaking up the fuel bed in order to ensure suitable porosity for an adequate distribution of air. For a similar reason separately operated slicing bars are placed between each retort unit, the small tuyere blocks on either side being stationary.

These various movements combine to keep the heavy mass of fuel constantly disintegrated under all conditions of working, a particularly necessary precaution with certain classes of heavy coking coals which otherwise would readily form into a solid mass through which the air necessary for complete combustion would be unable to find its way.

Separate movement is also provided for the extension plates on which the final stages of combustion take place, and the clinker when finally burnt out is automatically discharged over stationary ash plates.

Underfeed Stokers. The main principle of the wide range of apparatus manufactured by the Underfeed Stoker Co., Ltd.—apart from the "travelling grate" mentioned elsewhere—and included under the above heading, is that the raw coal is introduced on the underside of the fuel bed so that it is partly coked before it reaches the surface.

By this method of feeding, the volatiles which are given off by the green fuel have to penetrate an incandescent fire bed before entering the furnace, and in the process are sufficiently heated to ensure that when mixed with a correct proportion of air smokeless combustion will result.

The fuel is fed into a central trough extending from front to back of the furnace, and from the edges of this trough grate bars slope away to either side.

The actual feeding is accomplished either by means of a steam ram and push plate, extending over the length of the trough or, in smaller plants—as in Lancashire boilers and the like—by means of a worm or spiral.

With either arrangement the incoming fuel is eventually forced over the edges of the trough and moved gradually to the sides of the grate. In the case of the "E" type stoker this movement is assisted by means of the slightly reciprocating grate bars. Ash and clinker from the sides of the grates are removed periodically, either by hand or mechanical means.

Fig. 54 gives a general interior view of the "E" type stoker—one of the most popular of the series—as applied to a water tube boiler.

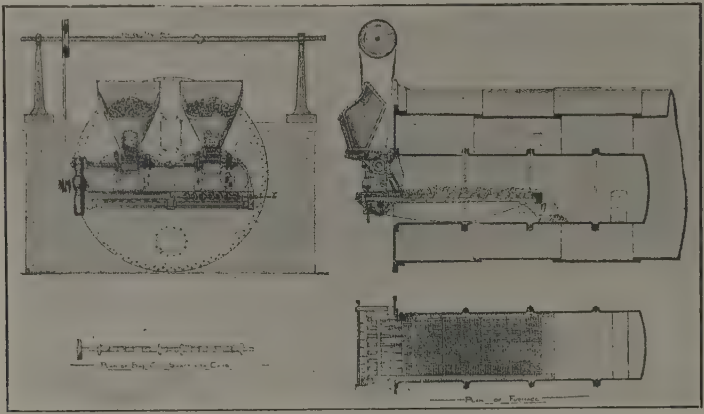


Fig. 55. "Bennis" Sprinkler Stoker and Compressed Air Furnace Applied to Lancashire Boiler

Sprinkler Stokers. The sprinkler stoker is one of the many types produced by Edward Bennis & Co., Ltd., and is more especially adapted for use with cylindrical boilers.

The principle adopted embodies one of the old golden rules for hand firing—"little and often"—without entailing the grave disadvantage of continually opening the furnace door which the adoption of such a rule makes necessary when a human operator is concerned.

Immediately outside the furnace front is fitted a small hopper from which a mechanical shovel automatically receives a predetermined and comparatively small amount of fuel.

The shovel arm is attached to a steel rocking shaft, and is forced back by cam movements against a long coiled spring, and released at definite and regular intervals by means of a tappet motion at the side. The effect of this sudden release is to scatter or "sprinkle" the contents of the shovel through the furnace exactly as a good fireman would do by hand. By varying the lift of the tappet, four degrees of "throw" are made possible in regular sequence, so that an even body of fire is maintained throughout the length of the furnace.

The fire grate used in conjunction with the "sprinkler" is made up of rectangular troughs about 4 in. wide which extend over the full length of the furnace and carry narrow fire bars in sections about 2 ft. long.

Air is forced into the troughs at the front end by means of small superheated steam injectors, and the bars are spaced to ensure even air distribution through the fuel bed.

The troughs are given a slow reciprocating movement of about 4 in. in all, the bars being pushed into the furnace together but withdrawn separately; the cam movements for this purpose being so arranged that as one trough is withdrawn those on either side of it remain stationary.

By this arrangement clinker is automatically broken up and carried to the back of the bars, where it passes into an ash dump and is removed as required.

THE TESTING OF PRIME MOVERS

By R. Royds, M.Sc., M.I.Mech.E.

LESSON V

INDICATORS FOR HIGH SPEED ENGINES

The ordinary piston and pencil indicators are useful instruments up to about 400 revolutions per minute, but with higher speeds the lag occasioned by the mass of the moving parts, and the relative slowness of the indicator oscillations, become so pronounced in their effects that no reliance can be placed on the diagrams. Modern designs have, therefore, sought to reduce the mass of the moving parts and to stiffen the spring, so that the indicator oscillations should be extremely rapid and of small amplitude. Frictional resistances have also been reduced in comparison with those obtained on ordinary indicators.

Diaphragm Optical Indicator. The manograph type of indicator was first introduced by the late Professor John Perry. There are various modifications of this type of indicator, but the usual lines of construction are shown diagrammatically in Fig. 26. In this arrangement a diaphragm C is connected to the engine cylinder through the cock B and pipe A. The variations of pressure deflect the diaphragm, and this motion is transferred to the point H on the back of the mirror M. A beam of light from an electric bulb E passes through a cross slit S and is caused to fall on the mirror M by the prism or mirror P. The tilting of the mirror M by the diaphragm causes the reflected spot of light to move vertically on the ground glass screen L. A small crank or eccentric K in phase with the engine crank is driven from the engine at the same speed, and actuates the point G at the back of the mirror M. This action alone tilts the mirror so that the spot of light moves horizontally over the screen L. F is a fixed point of support at the back of the mirror in the centre. The combined motions cause the spot of light to trace out a diagram of pressures on the screen L to a stroke base. If desired, a dark slide with photographic plate can be substituted in place of the slide L.

The main objections to this type of indicator are due to the heating of the diaphragm, lack of uniformity of the scale of pressure, the length of the pipe A with the change of engine clearance volume on opening the cock, and vibration of the indicator if supported from the engine. To keep down the heating effect on the diaphragm and to prevent "organ pipe" effect in the pipe A when running at high speeds, nickel gauze is sometimes inserted in the pipe A.

Hopkinson Optical Indicator. There has been a strong tendency in recent years to abandon the diaphragm arrangement for high speed work and, instead, to make use of a light piston and a strong beam spring supported externally to avoid heating of the spring. The Hopkinson optical indicator, built by Dobbie McInnes & Clyde, Ltd., was probably the first to adopt this arrangement. A sectional view is given in Fig. 27. The light hollow piston F connects by a clip to the beam spring D, the spring being supported at the two ends. The deflection of the beam is transmitted by the light link K to the pivot I on which the mirror H is mounted. The tilting of the mirror reflects a spot of light on to a ground glass screen and a photographic dark slide can be substituted for the screen to obtain a permanent record. The whole of the upper part of the indicator swivels about the ball bearings shown so that the motion is proportional to that of the engine piston. The mirror, being carried on this part, thus receives a motion so that the spot of light will travel across the screen and give a copy of the motion of the engine piston. Liners can be inserted in the cylinder with corresponding sizes of pistons

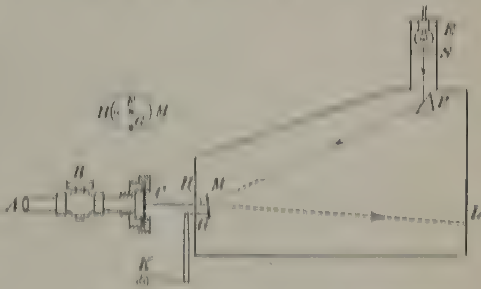


Fig. 26. Optical Indicator with Diaphragm

to alter the scale of the spring to suit the pressures being measured.

Objections are sometimes raised to the use of a beam spring supported at both ends because of frictional resistances at the points of support as the beam is deflected. This has caused other designers to prefer a cantilever spring. In the Burstall indicator the light barrel-shaped piston is connected to a cantilever type of spring bolted at one end to the body of the indicator. At the outer end of the spring a mirror is fixed so as to reflect a spot of light. The deflections of the spring, due to the load on the indicator piston, tilt the mirror and cause the beam of light, after falling on another rocking mirror, to move over a ground glass screen or photographic plate accordingly. This rocking mirror is given a motion proportional to that of the engine piston, and thus the combination of the motions of the two mirrors results in an indicator diagram being traced out by the spot of light on the screen.

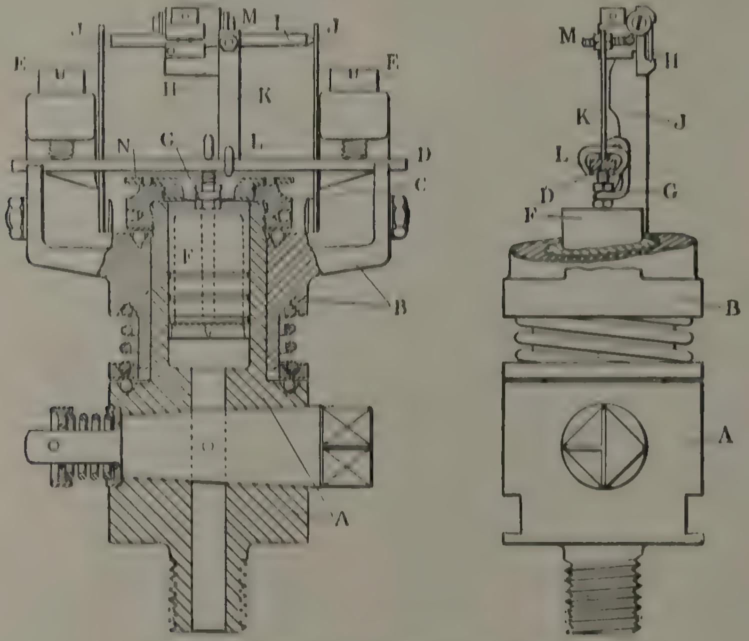


Fig. 27. Hopkinson Optical Indicator

Collins Micro-indicator. In the Collins micro indicator, built by the Cambridge Instrument Co., Ltd., the indicator piston is also connected to a cantilever spring, but no attempt is made to magnify the motion. On the end of the spring is a fine steel point or stylus arranged to mark on a piece of celluloid on the indicator drum. The drum receives a reciprocatory motion in the usual way and a record is thus obtained. The diagram, however, is very small, and a microscope with an internal scale is provided to magnify the diagram for purposes of measurement. It is claimed that there is very little frictional resistance between the marking point and the celluloid. The drum is so arranged that a number of diagrams may be taken on the same disc of celluloid. No doubt this indicator will be less affected by engine vibration than the other high speed indicators using optical methods of magnification.

R.A.E. Electrical Indicator. A high speed indicator developed at the Royal Aircraft Establishment is made by Messrs. Dobbie, McInnes & Clyde, Ltd., and is named the "Farnboro" Electric Indicator. It operates on a new principle as applied to indicators, and almost eliminates the effects of inertia and indicator oscillations. The primary operating parts of the indicator are shown diagrammatically in Fig. 28. A light contact disc on the underside is subjected to the cylinder pressure, while the other side of the disc has an adjustable air pressure acting on it, the air being supplied from an air bottle. The disc is in contact with either the seat A_1 or the seat A_2, and in either position the disc completes the primary circuit of an induction coil. If the disc valve is in contact with the seat A_1, say, the disc leaves this seat when the cylinder pressure rises to that of the air pressure on the top of the disc and at that moment breaks the primary circuit. Immediately the disc reaches the other seat A_2 the primary circuit is again made, but is broken again when the disc moves away at the moment the cylinder pressure again falls to that of the air pressure. The pressure of the air is adjusted by means of a throttling valve at gradually increasing pressures up to the maximum cylinder pressure. This same air pressure also acts on the piston of an indicator built on ordinary lines except that the "pencil" gear is insulated. This "pencil," or point, shown at *E* in Fig. 29, is thereby displaced parallel to the axis of the drum shown and its position is therefore a measure of the air pressure acting on the disc, and therefore indicates the cylinder pressure attained when the disc valve breaks the primary circuit. A paper is placed on the drum and rotates with the drum at the same speed as the engine or at half that speed. The frame of the indicator is a part of the secondary circuit and, at the moment the primary circuit is broken, the potential created in the secondary causes a spark to jump from the point *E* (which is in the secondary circuit) to the drum, and in its passage burns a small hole in the paper. Thus as the air pressure is varied a succession of sparks occur at different points on the paper until the whole diagram is completed. This

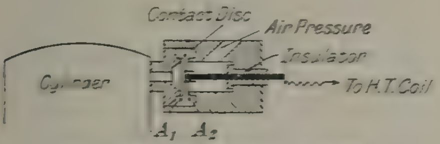


Fig. 28. Contact Disc in R.A.E. Indicator

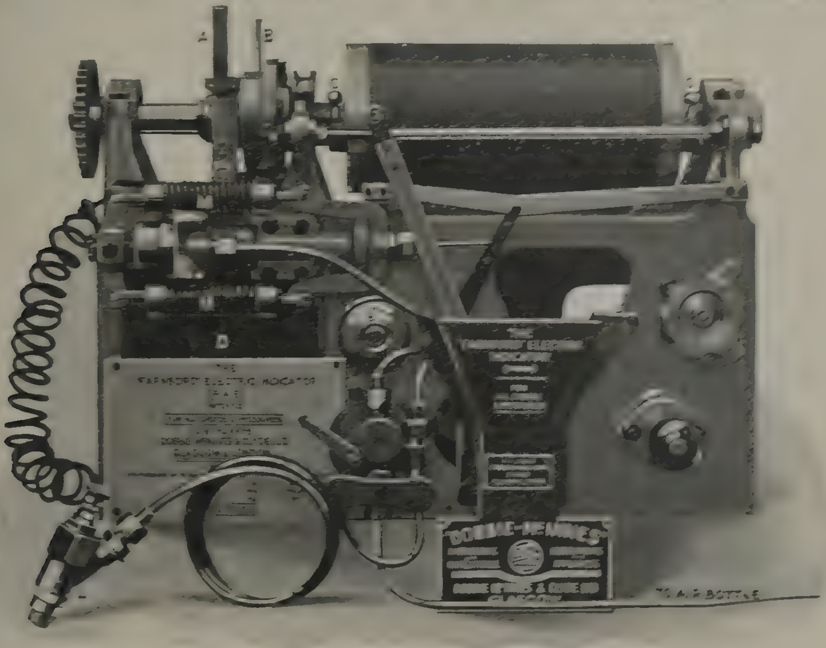


Fig. 29. R.A.E. "Farnboro" Indicator

diagram can be made to include points from any number of cycles varying from 100 to 1,000; the number of points on the diagram would then vary correspondingly between, say, 200 and 2,000.

The indicator drum is rotated in unison with the crank shaft of the engine and, therefore, the diagram is on a crank angle base. It is a comparatively simple matter to obtain from the rotary diagram the equivalent diagram on a piston stroke base.

Further details of some of these indicators may be obtained from the series of papers on "Indicators," read before the Institution of Mechanical Engineers in January, 1923.

The Gale[1] high speed indicator resembles the Farnboro indicator in so far that the diagram is obtained as a succession of points obtained from a succession of cycles, but differs in that the Gale indicator is wholly mechanical. The points at which the pressures are measured are selected by holes bored in two rotary cocks, or sleeves, one sleeve being driven from the engine in unison with the rotation of the engine crank and the other rotated by hand. That rotated by hand selects the point at which the pressure is measured, and the indicator piston, pencil, and spring, etc., move in accordance with the pressures exerted and gives a record on an ordinary type of drum.

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[1] *The Engineer*, 24th Sept., 1926.

PIONEERS OF ENGINEERING

By J. F. Corrigan, M.Sc., A.I.C.

LESSON XX

THE ATLANTIC TELEGRAPH

LORD KELVIN (1824–1907)

Born in Ulster in 1824, William Thompson (afterwards, Lord Kelvin) was both an engineering and an electrical pioneer. So numerous were his discoveries, and the improvements which he effected in engineering and electric technique, that it is impossible to mention more than the smallest fraction of them here. One important feature of Lord Kelvin's life, however, is the part which he played in the laying down of the first practically successful Atlantic telegraph.

From his earliest days, Lord Kelvin had been interested in the proposition of the Atlantic telegraph, and he followed with more than ordinary scientific interest the earlier attempts which were made to lay it. Before Kelvin's participation in the Atlantic telegraph scheme, one of the great difficulties which confronted the cable layers was that of devising a method of paying out the cable in a uniform manner as the ship proceeded in its course. Kelvin, however, devised a method of making the cable pay itself out automatically. If any break occurred during the paying out operations, the end of the cable was automatically gripped, thus obviating the necessity of having to spend many laborious days in searching for the lost cable.

Kelvin's method of sounding the ocean bed was of the utmost assistance to the early cable layers, as was also the method he invented for coiling the cable in the ship in the minimum of space.

To him also are due a special mirror reflecting galvanometer used in long distance telegraph work, and the now greatly used syphon recorder.

Kelvin's researches, assistance, and advice in the Atlantic telegraph scheme ultimately resulted in the successful completion of the cable in 1866, and in recognition of his work in that direction, he received the honour of a knighthood, becoming Sir William Thompson.

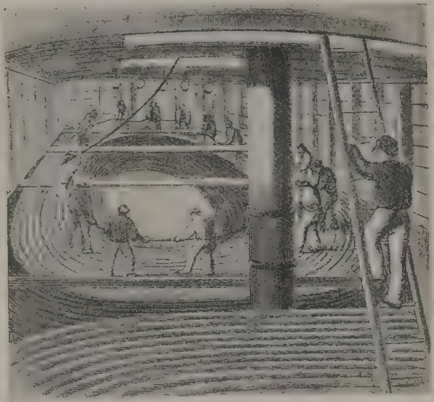
In addition to his work on the Atlantic

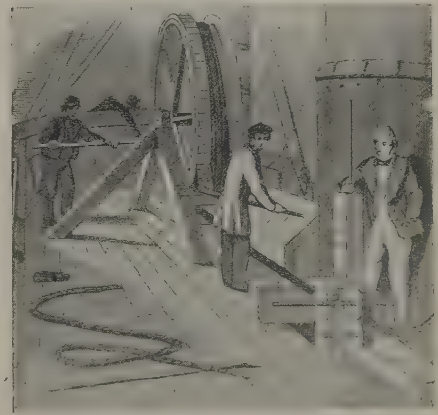
telegraph, Kelvin closely followed up and advanced the work of Joule and Carnot on the relations of heat and energy. He enunciated the principle of the dissipation of energy, a law which was at first disputed but which was afterwards fully accepted. As a successor to Joule and Carnot, Lord Kelvin has incurred the debt of the engineering world. His work on the nature of heat and energy, and his insistence on the validity of the doctrines of his predecessors, resulted in engineering theorists and designers gaining a firmer and a clearer perception of the problems which were set before them.

LORD KELVIN

Kelvin was educated at Cambridge. He received his degree from that University, and at the early age of twenty-two he was appointed to the Chair of Natural Philosophy at Glasgow University. Queen Victoria raised him to the peerage in 1892, the year in which he occupied the presidency of the Royal Society. Between these two events which took place at the extremes of his life is to be found a long period of ceaseless work and activities on the part of Kelvin in one scientific or engineering field or another.

Lord Kelvin died in 1907. He was buried in Westminster Abbey with simple yet impressive ceremony, and with honours which were most deservedly due to him, if only on account of his major participation in an attempt to link together two powerful nations in the bonds of peace and friendship by the material and symbolic power of a slender telegraph line.


AN EARLY METHOD OF COILING THE CABLE IN A SHIP'S HOLD


EARLY CABLE-LAYING MACHINERY

BLAST FURNACE PRACTICE

By Percy C. R. Kingscott, D.I.C., A.R.C Sc., F.I.C., B.Sc.(Lond.)

LESSON IX

MATERIALS USED IN IRON SMELTING

Ores

Although iron comprises about 4·4 per cent of the crust of the earth, it very rarely occurs native, with the exception of that in the form of meteorites. Iron has a strong affinity for oxygen; thus, the most important source of iron, from the point of view of the smelter, is that of iron oxide. It also occurs as carbonate and silicate of iron, both of which compounds are used in the blast furnace.

An ore consists of the compound of iron associated with a gangue of earthy material, and in the action of smelting, the iron compound is broken up, liberating metallic iron. The gangue combines with the added flux to form the very essential slag.

This gangue usually consists of silica, alumina, lime, magnesia, oxides of manganese, compounds of sulphur, alkalis, etc., in varying proportions. Sometimes the proportions of silica, or acid constituent, and lime and magnesia, or basic constituents, are such as to constitute a self-fluxing mixture, but generally there is a deficiency of basic materials, and this must be made up by the addition of basic flux in order to form a workable slag. In some districts the gangue is highly calcareous and needs the addition of acid or siliceous compounds in order to flux it.

The chief types of ore are as follows—

Magnetite (Fe_3O_4). This ore contains, when pure, 72·4 per cent of iron, and is thus named on account of its property of being attracted by a magnet. It is a very dense ore, and is not easily reduced in the blast furnace. Rolling mill scale is similar in composition.

Hematite (Fe_2O_3). This ore contains, in the pure state, 70 per cent of iron and is, as its name implies, of a red colour. Red iron rust is of identical composition. Hematite ores occur in two main divisions of commercial value, viz.—

(*a*) Ore low in phosphorus content suitable for acid steel manufacture.

(*b*) Ores containing larger amounts of phosphorus suitable for basic steel manufacture or ordinary foundry work.

Brown Ore (Fe_2O_3,nH_2O). These ores are hydrated sesqui-oxides of iron, differing from hematite ore only by the presence of combined water. There are several varieties, the chief of which is **Limonite,** or $2Fe_2O_3, 3H_2O$ containing, when pure, 59·9 per cent of iron. When heated to a sufficiently high temperature it breaks down to the red hematite giving off the combined water. Limonite is of a yellow colour, and gives a yellow streak on porcelain.

Other ores of varying degrees of hydration give streaks varying from yellow to red.

Siderite ($FeCO_3$). Siderite or spathic ore contains, when pure, 48·3 per cent of iron and 37·9 per cent of CO_2. It is rarely used direct in smelting, but undergoes preliminary calcination in order to expel the CO_2 and other impurities leaving the red ore. It is sometimes associated with coal measures as black band ore.

Silicates. When found naturally, these ores generally contain small amounts of iron. They are generally obtained as slags or cinders from puddling and reheating furnaces. They usually contain about 55 per cent of iron and 20 to 30 per cent of silica. These cinders are very variable in composition, but are valuable in increasing the slag volume or in decreasing the ratio of alumina to silica in the slag when aluminous ores are used.

Composition of Ores

The value of an ore depends largely upon the kind of iron to be produced, and thus it must conform to a certain range of composition. Amongst the materials present are some which, in the process of smelting, combine with the molten iron, giving it certain characteristics. The chief among these elements are silicon, phosphorus, sulphur, and manganese.

Silicon is derived from silica, but that entering the iron probably comes chiefly from the ash in the fuel. Silica in ores is a diluent, but is necessary in admixture with limey ores.

Phosphorus is not controlled by the smelter. It is thus essential that the phosphorus content of the ore must not exceed that permissible in the corresponding quantity of iron produced, for the latter absorbs all the phosphorus present in the burden. Phosphorus is generally present as Apatite or calcium phosphate.

Sulphur is a very deleterious element in ores and, if present in large quantities, should be removed by preliminary roasting. Its presence requires the use of extra flux and fuel. Sulphur occurs chiefly as sulphide of iron, but is also to be found as sulphates of basic materials.

Manganese is more difficult to extract from its compounds than is iron, and the quantity reduced depends upon several factors in the furnace hearth. Under normal conditions, from 50 to 75 per cent of the manganese present in the ore is reduced and enters the molten iron. It prevents the combination of sulphur with the iron, and ores containing 5 to 10 per cent of manganese are sometimes deliberately mixed with other ores in which the sulphur content is too high.

Other components of the ore, such as alumina, lime, magnesia, and the alkalis do not affect the ore from the standpoint discussed above, as they do not enter the iron.

Accessibility. The accessibility of an ore is obviously of great importance as regards transport to the fuel or vice versa, and also the distance from the iron consumers or ports. However, the qualifications affecting the value of an ore are interdependent, for it is possible for a rich ore to be associated with deleterious elements to such an extent as to render it valueless, and yet an ore containing a much smaller percentage of iron may be exceedingly valuable owing to its general accessibility or because it is calcareous or even self-fluxing.

VALUATION OF ORES

Pig-iron, as stated above, is always associated with other elements, and generally contains from 92 to 95 per cent of pure iron, whether produced from a rich or a poor ore. It has been found that this fixed amount of iron takes a fairly constant amount of carbon for reduction, impregnation and melting; hence the valuation of an ore depends largely upon the treatment of the slag. The quantity of the latter depends upon the amount and properties of the gangue, for upon these depend the quantity of flux to be added. The quantity of fuel required to melt the slag is also known with a fair amount of accuracy.

Providing that the ores are of the same class and that they are used in the production of similar kinds of pig-iron, the following method of valuation given by Forsythe can be used—

Analysis of Ore.

Fe	SiO_2	CaO	MgO	Al_2O_3	MnO	P	S
43	15·7	1·2	·5	3·1	·5	·04	·04

Flux.

52 per cent available base or *efficiency* of the limestone.

55 per cent slag forming material.

Fuel.

79 per cent available carbon.

Assume 25 per cent limestone required to flux the ash.

Slag.

Ratio bases to acid, 1·3 : 1.

Pig-iron.

Si, 2 per cent; Fe, 94 per cent.

The amount of ore required, assuming no smelting loss, to produce 1 ton of pig-iron =

$$\frac{94}{43} = 2{\cdot}18 \text{ tons.}$$

The amount of silica reduced to produce 2 per cent Si in 1 ton of pig = ·043 tons.

Hence, determining the quantity of limestone required per 100 units of ore—

SiO_2 in ore . =	15·7	CaO in ore	1·2
SiO_2 reduced to Si =	1·9	MgO ,,	·5
		$\frac{1}{3}$ MnO ,,	·17
SiO_2 to flux . =	13·8		1·87

Bases required to flux the silica are given by—

13·8 × 1·3	17·94
Already present . . .	1·87
Bases from flux . . .	16·07

Amount of limestone flux

$$= \frac{16{\cdot}07}{{\cdot}52} = 30{\cdot}9 \text{ per cent of the ore.}$$

Hence, limestone per ton of pig

$$= 2{\cdot}18 \times {\cdot}309 \text{ tons}$$
$$= {\cdot}6736 \text{ tons.}$$

The weight of slag from the ore per ton of pig

= (·138 + ·031 + ·0187) × 2·18 = ·409 tons
Slag from flux = ·6736 × ·55 = ·370 tons
Total ·779 tons

Assuming that the slag requires 25 per cent its weight of carbon to melt it—

	Tons.
Carbon to melt the slag = ·779 × ·25	= ·195
According to Gordon, carbon needed to reduce, impregnate, and melt pig-iron with 1 per cent Si, and blast at 1,000°–1,200° F.	= ·660
Carbon for extra 1 per cent Si . . .	= ·050
	·905

	Tons.
Coke required per ton of pig = $\frac{\cdot 905}{\cdot 79}$	= 1·145
Allow 5 per cent extra for breeze	057
Total coke	1·202

The fuel requires 25 per cent limestone to flux the ash, hence, total limestone is—

	Tons.
Required by ore .	. ·6736
Required by coke ash	. ·3005
Total	·9741

Knowing the prices of the above materials, it is possible to compare the values of different ores assuming constant overhead charges. Similarly, the cost of production of pig-iron can be calculated provided that wages, repairs, fixed charges, and other factors are known.

METALLURGY FOR ENGINEERS

By James Knowles

LESSON VII

THE BESSEMER PROCESS

The steel making processes thus far described have been those concerned with the manufacture of steel for cutlery, springs, engineers' tools and high-class cutting purposes generally. In these the carbon contents usually run from 0·60 to 1·60 per cent; they are produced in relatively small quantities and, in many cases, may be quoted and sold by the pound. Their original source is the black magnetite ore of Sweden, whose purity is reflected in the finished products.

There still remains that other great class, viz., structural steels—to be considered. These are made by either the Bessemer or open hearth processes, from which tremendous outputs are obtained, whilst selling prices are much less. Carbon contents too, as a general rule, run considerably lower, say, from 0·10 per cent upwards, although, if desired, good steel of 2·0 per cent carbon can be made. The source of this class is red or brown hematite.

The Bessemer process in Great Britain has now been almost superseded by the open hearth and, in view of the rapid development of electric methods of steel melting, a corresponding fate would appear to threaten the crucible—at least so far as special alloy steels are concerned.

It will be remembered that this last mentioned process really consists of a mixing and melting operation; the Bessemer and open hearth principles, on the other hand, are essentially chemical in their conception and depend for success on the removal, by oxidation, of certain elements in the pig-irons employed.

In the former a large volume of air is blown at high pressure through a bath or molten pig-iron and whilst, in an acid lined vessel, the sulphur and phosphorus cannot be eliminated by these means, the oxygen so supplied unites with the carbon, silicon, and manganese. The oxides of silicon and manganese, being insoluble, form a slag on the surface of the metal whilst the carbon passes off from the bath in a gaseous state, almost entirely as CO. These reactions, being accompanied by a considerable evolution of heat, serve to create and maintain a bath temperature adequate for subsequent casting operations—either into ingot or steel foundry moulds. Hence, it will be seen that, commencing with pig-iron practically free from sulphur and phosphorus, an ultimate product chemically similar to wrought iron can be obtained.

In 1856 Henry Bessemer read a paper before the British Association in which he said he proposed to make pure iron from pig in 20 minutes and in 10 ton lots at a time. This created great amusement—no metallurgist dared have made such a startling proposition, but Bessemer was not a metallurgist, he was a solicitor. A similar paper given before the Institute of Civil Engineers in 1859 was also received with scepticism, which appeared well founded when large scale operations on the foregoing lines were undertaken.

A small percentage (about 0·2) of the oxygen was found to remain as ferrous oxide, dissolved in the blown metal, causing violent boiling in the moulds and unsound ingots. This oxide, on solidification, forms an enveloping film around the crystals of iron and softens on reheating, causing the whole mass to fall to pieces. Such material is said to be "red short," and is utterly worthless.

The saviour of the process was R. Mushet, who discovered that the addition of metallic manganese to the blown metal would, by the formation of insoluble manganese oxide, carry off this oxygen to the slag and so yield sound ingots.

This necessary manganese is added in one, or both, of two forms, viz., ferro-manganese and spiegel, of which the following are typical analyses—

	Ferro-manganese		*Spiegels*	
Carbon	6·50	5·50	5·00	4·50
Silicon	0·90	0·70	1·00	1·00
Manganese	80·00	50·00	20·00	15·00
Sulphur	0·02	0·02	0·01	0·01
Phosphorus	0·18	0·18	0·09	0·09
Iron	12·40	43·60	73·90	79·40

All these are alloys of iron, manganese, and carbon, the spiegels containing a much smaller percentage of manganese. From these analyses it will readily be seen that the addition of a determined proportion of manganese, by careful selection and calculation, can also be accompanied by a desired addition of carbon, so as to make up the bath percentages of both these elements to required figures. Broadly speaking, if a low carbon heat is desired ferro-manganese is used, and, since a relatively small amount will be needed, this can be added cold. On the other hand, if a rail or spring heat is to be worked, a considerable quantity of spiegel will be used and this, to avoid a chilling effect, will need to be charged after being melted in a small auxiliary cupola.

A plentiful use of manganese is of the utmost importance in the manufacture of Bessemer steel—a fact which was not sufficiently recognized in the early days of its history—and an excess of about 1·0 per cent should be left in the finished steel to ensure soundness and working qualities.

The Bessemer principle is applied in several ways, all of which should be sharply differentiated. They may be classified as follows—

1. Acid, 2. Basic, or (*a*) English, (*b*) Swedish.

Since, by the oxidation of the carbon, silicon, and manganese, the heat required for the operation is developed, these elements may be considered as fuels, self-contained in the pig-iron. In this respect carbon is the least important, as the products of its combustion escape as CO gas and burn to CO_2 *outside* the vessel. Manganese plays a somewhat more useful rôle, but it is the silicon which is *the* great heat producing element. Hence, in selecting charges for the Bessemer process, an average silicon content of about 2·0 to 2·5 per cent should be aimed at. Above this the charge would work hot, and below a liability to "coldness" would exist. Dr. J. O. Arnold recommended a figure of 2·3 per cent as the best for English practice. Sulphur and phosphorus must obviously be kept as low as possible.

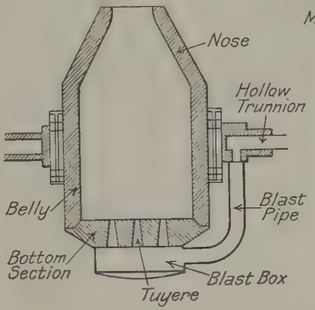


FIG. 7

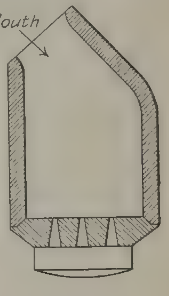


FIG. 8

The pig iron, or irons, selected can be melted in a cupola; transferred in ladles direct from the blast furnace; or conveyed thence to a mixer. A mixer is essentially a large gannister lined vessel designed to act as a reservoir for the molten iron which is poured off as, and when, required for conversion. By these means the iron from several blast furnaces can be mixed and any irregularity in quality in the product of an individual furnace neutralized. A considerable saving is thus effected since remelting of the pig-iron is not necessary. Obviously, if a cupola is worked in conjunction with the Bessemer plant, corrections must be made for melting changes in the composition of the iron.

The Converter. The Bessemer converter, Figs. 7 and 8, is an iron or mild steel vessel, lined with rammed gannister, and is roughly pear shaped. Converters may be concentric or eccentric, according as the sides are equally inclined to the axis or the nose designed with one straight and one curved side. The latter is the more familar form. It is customary to consider a converter as being built up in three parts—the bottom section, the belly and the nose, all being bolted and keyed together and made detachable for ease of renewal and repair.

The whole is mounted on trunnions, supported on pedestals, so that a semi-rotary, or rotary motion can be effected, either by means of a rack and pinion actuated by a double-acting hydraulic ram; or by worm and worm wheel. A number of fire-clay tuyeres are contained in the centre of the bottom section, through which the air is blown at a pressure of from 20 to 25 lb. per sq. in., dependent on the depth of the charge. Below the tuyeres is the blast box, or air chamber, which receives the air from the blast main by way of a pipe connection with one of the trunnions, which is made hollow for the purpose, and is directly connected with the blast main.

The Blow. The "blow" occupies from 15 to 25 minutes and is technically divided into three stages: (*a*) the preliminary, (*b*) the "boil," and (*c*) the finishing. All these are clearly marked by variations in the fume and flame issuing from the mouth of the converter, and an ability to read and interpret these variations is the vital qualification for the position of Bessemer "blower."

The converter, when ready for the reception of the charge, is rotated to a horizontal position and the molten metal poured in from a ladle. The level of the charge must not be allowed to reach the tuyeres. The blast is then admitted and the converter turned back to the vertical, the pressure of the blast being sufficient to prevent the metal running down into the tuyeres and to force the air through the charge.

(*a*) Of the three elements mentioned in this connection, silicon is the most readily oxidized and is therefore attacked first. So great is its affinity for oxygen that until it has been almost completely eliminated the oxidation of the carbon cannot proceed. During the preliminary stage the removal of the silicon and manganese is almost totally effected. A dense brown fume, showers of sparks and a short flame first issue from the mouth of the converter, but, as the oxidation of the silicon proceeds, with consequent rise in temperature, the size and steadiness of the flame increase, and its luminosity becomes brilliantly white.

(*b*) The violence of the reaction, the heat and fluidity of the bath continue to increase until, some five minutes after the commencement of the blow, the "boil" occurs. This stage is evinced by the ejection of molten splashes of slag and marks the tremendous agitation of the bath by the escaping CO gas consequent on decarburization. It is during this phase that the blower is able to estimate the heat of the charge, which is rectified by additions of steel scrap or ferro-silicon according as to whether it is adjudged too hot or too cold.

(*c*) The flame gradually subsides until finally it "drops" altogether denoting almost complete decarburization. This is the signal for the cessation of blowing, and immediately it is observed the vessel is turned down and the blast shut off. Any prolongation of the blow would have the damaging effect of oxidizing the iron. The additions of manganese are then made by means of an overhead shoot with slip bottom, the contents of the converter poured into a waiting ladle and the ingots cast.

Sulphur and phosphorus percentages always rise slightly during the blow owing to the concentration of the bath, and the following tables show the net results: (1) being the chemical composition of the cupola metal, and (2) the blown metal before the addition of manganese—

	(1)	(2)
Combined carbon	0·50	0·04
Graphite	3·5	nil
Silicon	2·50	0·03
Manganese	0·40	nil
Sulphur	0·05	0·06
Phosphorus	0·05	0·06

POWER TRANSMISSION BY BELTING

By W. G. Dunkley, B.Sc.

LESSON IX

WORKED EXAMPLES. SOME SIMPLE RULES

In the previous examples which have been given we desired to find the horse-power, given the pulley, speed, and size of belt, and to find the size of belt given, the horse-power, pulley, and speed. Now let us consider the following case—

Example 16. The machines in a shop are driven by means of a countershaft. We desire to drive the countershaft at 300 r.p.m. from a 60 h.p. motor which runs at 900 r.p.m. Determine suitable pulleys and belt. Distance from motor to countershaft 12 ft.

Now you see we have a wider choice of conditions. We first notice that we have a speed reduction of 3 to 1. If we make our motor pulley 10 in., the countershaft pulley would be 30 in.; if we make the motor pulley 15 in., the countershaft pulley would be 45 in. and so on.

Now the larger the pulleys the greater the belt speed and, consequently, the effective driving load for a given horse-power is less. Therefore a small belt will transmit the power. So far as the belt is concerned, it is economical to provide the pulleys as large as possible. This condition is generally decided by the size of the pulley on the countershaft being limited to a certain maximum value, determined by the distances of the countershaft from the wall or supporting girder. If the pulleys are not subject to such limiting conditions the belt speed then becomes the deciding factor. At high speeds the centrifugal force increases very rapidly, and beyond a certain speed there is actually a falling off in horse-power as the speed increases. For single belts, cemented, this speed is about 5,500 ft. per minute; for laced single belt and double cemented belt the speed is about 4,800 ft. per minute, and for a double belt, laced, it is about 4,500 ft. per minute.

Suppose we decide to use a double belt with a cemented joint and a belt speed of, say, about 4,000 ft. per minute. Now let us find what our pulleys will be. Let D = diameter of motor pulley in inches.

$$\text{Then } \frac{D}{12} \times \pi \times 900 = 4000.$$

$$D = \frac{4000 \times 12}{\pi \times 900} = 17 \text{ in.}$$

The pulley on the countershaft would then be 51 in. diameter. These are somewhat odd dimensions, and we would probably consider 16 and 48, or 18 and 54 in. Suppose we adopt 18 and 54 in. diameter pulleys. Now applying our procedure—

1. Setting down the pulleys to scale at the centre distance of 12 ft. we find that the angle of lap on the motor pulley is 160°.

$$\text{Then log. } \frac{T}{t} = 0{\cdot}007578 \times 0{\cdot}4 \times 160$$

$$,, \quad \frac{T}{t} = 3{\cdot}06$$

$$2.\ P = \frac{H \times 126{,}000}{D \times N} = \frac{60 \times 126{,}000}{18 \times 900}$$

$$= 466 \text{ lb.}$$

$$3.\ T = \frac{\frac{T}{t} \times P}{\left(\frac{T}{t} - 1\right)} = \frac{3{\cdot}06}{2{\cdot}06} \times 466$$

$$= 700 \text{ lb.}$$

4. Belt velocity

$$V = \frac{18}{12} \times \pi \times 900 = 4250 \text{ ft. per min.}$$

Stress S due to centrifugal force

$$= \frac{V^2}{230{,}000} = \frac{4250 \times 4250}{230{,}000}$$

$$= 80 \text{ lb./sq. in.}$$

5. We decided to use a double belt with cemented joint. Then allowable stress = 300 lb./sq. in. Allowing for centrifugal force we must use $300 - 80 = 220$ lb./sq. in. Then $x \times \frac{3}{8} \times 220 = T = 700$

$$x = \frac{700 \times 8}{220 \times 3} = 8{\cdot}5 \text{ in.}$$

Example 17. Suppose in the above installation we desire to know the total pull on the shaft in order to determine the bending moment on the countershaft, due to the belt pull.

The total pull will be approximately equal to $T + t$. This is not quite correct since the two sides of the belt are not parallel. We must then find t.

$$t = T - P = 700 - 466 = 234 \text{ lb.}$$

$$\text{Belt pull on shaft} = 700 + 234 = 934 \text{ lb.}$$

You should notice that this is about twice the effective driving load.

Simple Rules for Belt Calculations

We have seen that in our belt calculation, the varying value of the angle of lap, the necessity of allowing for centrifugal force, and the possibility of desiring to make some change in the value of the coefficient of friction, make it desirable to investigate the problem in the manner shown.

Simple rules are frequently given, however, and these are useful and applicable for rough calculation and estimates, provided the restrictions underlying these rules are appreciated. These rules are generally applicable to an angle of lap of 180°, and should be used with caution in general cases. The following rules are given by Messrs. Tullis—

1. Every inch width of good double leather belting travelling at 500 ft. per min. while working on pulley 24 in. diameter will transmit 1 h.p., provided the arc of contact is 180°.

2. Every inch width of good single leather belting travelling at 800 ft. per min. will transmit 1 h.p. on pulley 24 in. diameter at 180° arc of contact.

The horse-power will vary proportionately with the speed up to speeds of 3,000 ft. per min.; above this speed the allowance necessary for centrifugal force will reduce the horse-power.

In the table on this page the values are given by the Gandy Belt Manufacturing Co.

Effective Pull per Inch Width of Belt. It will be instructive to compare the effective pull per inch width obtained in the calculations we have made. Starting from Example 10, we find—

Effective pull per inch width of belt = 42·5 lb., lap 175°, single belt; 35·5 lb. lap 130°, single belt; 41·5 lb., lap 170°, single belt; 62 lb., lap 170°, double belt. The results where centrifugal force is allowed for are not considered.

For angle of lap 180° and $\mu = 0{\cdot}4$ the following values of effective driving load per inch of width of belt may be used—

Single leather belt (cemented) 50 lb.
,, ,, ,, (laced) 40 lb.
Double ,, ,, (cemented) 80 lb.
,, ,, ,, (laced) 65 lb.

Example 18. A machine is supplied equipped with a pulley 18 in. diameter, $4\frac{1}{2}$ in. wide, speed of pulley 300 r.p.m. A single laced belt is to be used. Make a rough estimate of the size of motor required to drive the machine.

Balata Belting	Leather Belting	H.P. per inch width at 100 ft. per min.
3-ply	Light Single	0·12
4-ply	Ordinary Single	0·15
5-ply	—	0·18
6-ply	Light Double	0·21
7-ply	—	0·24
8-ply	Heavy Double	0·27
9-ply	—	0·30
10-ply	Treble	0·33

$$\text{Belt speed} = \frac{18}{12} \times \pi \times 300 = 1410 \text{ ft. per min.}$$

Then allowing 40 lb. per inch of width and assuming a belt 4 in. wide,

$$\text{H.P.} = \frac{40 \times 4 \times 1410}{33{,}000}$$
$$= 7.$$

Allowing for overload, a 10 h.p. motor might be suitable.

Study. (*a*) Assume a double-laced belt in the above example. *Ans.* h.p. 10·5, motor, 15 h.p.

(*b*) Find belt required to drive a machine from a 20-h.p. motor, pulley 15 in. diameter, 5 in. wide, speed 900 r.p.m.

WORKS ORGANIZATION AND MANAGEMENT

By E. W. Workman, B.Sc., A.M.I.E.E., etc.

LESSON IX

THE PURCHASING DEPARTMENT

Functions of the Purchasing Department. In many works the duties of the purchasing department are not organized separately, but are amalgamated with those of the store-keeping department under one "Purchasing Agent and Head Store-keeper." This organization does not essentially alter the nature of the work to be done, and it will be more convenient to consider the work in different lessons as though the departments were separate. Good purchasing is very much more of a science than is usually recognized by the average individual. Practically anybody can merely place the order for goods—but "merely placing the order" does not constitute *good* purchasing. For purchasing to be really efficient, a wide knowledge of markets and prices is required together with fair technical understanding, and the possession of sound judgment as to relative values with varying conditions. If the cost of material forms a large proportion of the total cost—and it very often is 50 to 75 per cent—the effect of good or bad purchasing will be very apparent in the final price. All information necessary for efficient buying is systematically recorded to avoid waste of time and to ensure the best purchasing. Such records will include details of suppliers, complaints of material received, late deliveries, and tests made. Such information will be of great value when the question of repeat orders arises.

The Head of the Department. The official in charge of this department is usually described as the purchasing agent, and the position is usually a very responsible one. The qualifications are varied, as the purchasing agent must know the markets and the probable trend of them, and also the details of such matters as packing, freight, discounts, and insurance. In addition he must be able to judge of the technical suitability of the material he is buying.

Qualifications. For this position, more than for any other, it is necessary to select a man whose honesty is above all suspicion. There is no other position where the honesty of the official may be so often tempted to underhand transactions for personal gain. There are always many suppliers who are eager to offer some bribe to obtain an order—and the system of bribery may be of such a gradual and unostentatious nature that the purchasing agent may commence to accept gratuities which place him under obligations without his realizing what is happening. In order to protect the official as much as possible, the policy and organization of the firm should be well defined on the following matters—

1. The agent himself must be of high moral standing, and also capable of making firm decisions and of taking a strong stand.

2. It should be generally understood both outside and inside the factory that no gratuities are allowed to be received by the purchasing agent. It is illegal for an agent to receive a commission or other bribe and not disclose it to his employer, but more than this is required, and the whole known policy of the firm must be set against bribery in any form.

3. If proper comparative records are kept of the estimates of different suppliers and of all complaints made in regard to the quality of any material bought, and if the incoming material is inspected by a different department to that making the purchase, the difficulties of making any underhand transactions will be considerably increased.

Mental Characteristics. In addition to the supreme qualification of honesty, a purchasing agent should have several mental characteristics well developed. He should be "go-ahead," and always on the look out for improvements, not merely contenting himself with always ordering material exactly as specified. It is he who is in touch with outside interests, and it is he who should often be in a position to draw the attention of those concerned in his own firm to improvements adopted or suggested by the suppliers with whom he is in touch.

The extent to which his judgment is reliable will have a big effect in the prosperity of the firm. Sometimes it is advisable to lay in large stocks even at high prices because of the uncertainty as to the obtaining of future supplies, due to labour or political troubles. At other times it may be safe to reduce stocks to a minimum and later buy at much reduced prices. Such decisions require wide knowledge and judgment as to the trend of markets and general conditions.

Requisitioning. The purchasing department only buys goods for which it has received requisitions from other departments—for itself it requires no purchases. These requisitions should be on a standard form, originate in definite departments, and bear definite signatures authorizing the purchases. Whether the department issuing the requisitions should be the drawing office, the production department, or some other section will depend on the organization adopted for these departments.

Standards. Purchases fall naturally into two classifications: (1) special purchases, and (2) standard purchases. The first group require very careful specification by a technical department and, wherever possible, standard articles should be bought instead. The advantages of buying standard articles rather than special ones are as follows—

1. The number of articles used in the factory is reduced.
2. The reduction of quantity and variety means simplified store-keeping.
3. The possibility of ordering larger quantities (often at a cheaper rate), because the smaller variety means a larger consumption of each kind.

Without careful watching, the variety of articles carried in stock is sure to get unnecessarily large and correspondingly expensive.

Quantity Ordered. The quantity to be ordered at a time forms an important question. Each requisition should not be dealt with separately, but should be treated in conjunction with the demands shown on other requisitions, and also with probable future requirements. To be able to order in large quantities gives a number of clear advantages, such as—

1. Possibility of obtaining lower prices for large orders.
2. Lower percentage of carriage charges.
3. Lessened probability of running out of stock.
4. Probability of more regular deliveries and generally better attention.

The most economical course will often be to place a large order for the regular delivery of small quantities. By this means all the above advantages are obtained and, at the same time, the outlay of capital is reduced.

Delivery Dates. If only it could be depended on, the best time for the delivery of goods would be just as they were required for manufacturing or assembly purposes. But the risk of delay and breakdowns is too great to make this date the one usually asked for. At the same time, it must be recognized that to buy goods too long in advance may be expensive. The main items of extra expense involved in early purchasing are as follows—

1. Occupation of storage room, and the possible consequent necessity of increasing the storage accommodation.
2. Extra insurance premiums required to cover the risk of loss of the increased value of stores.
3. Greater loss in depreciation through extra handling, the effects of time and possibly obsolescence.
4. The expenses of having a larger outlay of capital.
5. The increased clerical and handling expenses involved by the carrying of a larger stock.

These extra expenses may, however, be much more than counterbalanced by the saving effected in buying early when prices are rising. Apart from any such special conditions the required dates of delivery can be best determined from the records of the production department, whose special function is to plan the dates when parts should be manufactured or bought. If there is any doubt on the subject, it will always be better to be on the safe side and arrange for deliveries to be too early rather than too late, as very heavy expenses are likely to be incurred if some important part is not to hand when required. For a firm to build up a reputation for keeping their promises of delivery of their finished products, they must ensure that their suppliers of raw material similarly keep their promised delivery dates, and "too early" is better than "too late."

Order Forms. It is of great advantage to have a recognized order form, and for it to be understood that no orders will be recognized as official unless they are given on the authorized order form. Besides eliminating unofficial ordering, the proper drawing up of such a form will ensure that mistakes and delays do not occur through lack of sufficient particulars. In particular, provision should be made for stating the delivery date required, the place of delivery, the price, the method of payment, and the detailed specification.

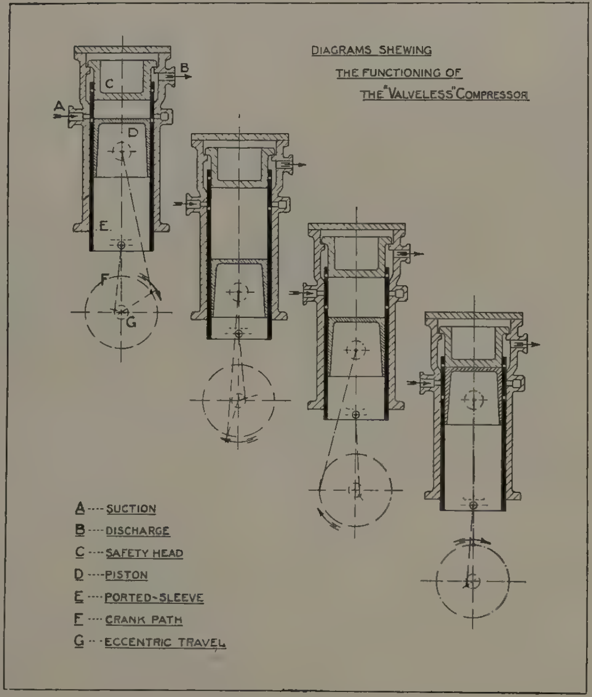


FIG. 17. OPERATION OF SLEEVE-VALVE MACHINE

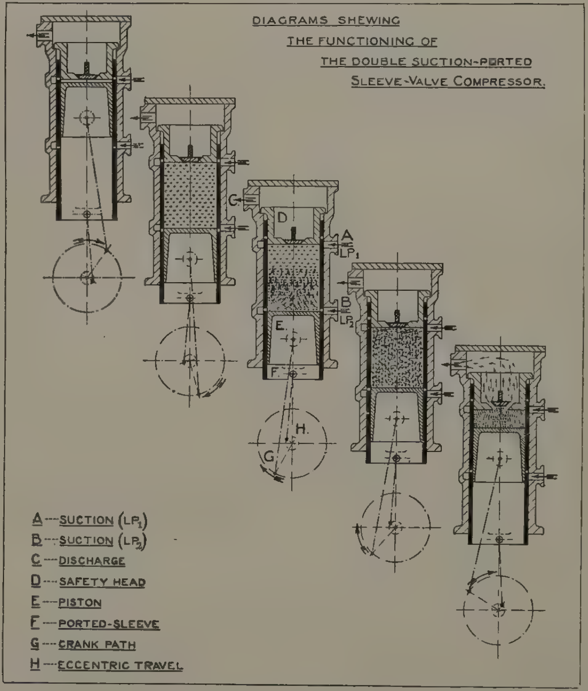


FIG. 18. OPERATION OF SLEEVE-VALVE MACHINE ON TWO-PRESSURE SYSTEM

AMMONIA COMPRESSORS WITH SLEEVE VALVES

(See Article on " Refrigerating Machinery," page 1128)

MATHEMATICS FOR ENGINEERS

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LESSON XXIII

DIFFERENTIATION—(*contd.*)

Graphical Aspect of Differentiation. In the last lesson we found the rule for differentiating any expression which is composed of powers. We already know the value of graphs, enabling us, as they do, to see things at a glance, and it will be of service in getting a clearer idea of the meaning of the algebra used in the last lesson, as well as leading to some of the important applications, to follow out the process of differentiation on a graph.

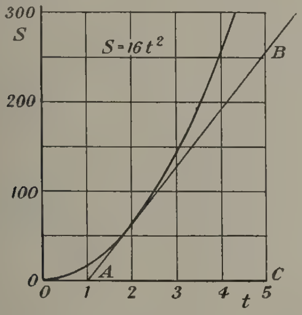


Fig. 91

As a first instance we will use the formula $s = 16t^2$, the graph of which is drawn in Fig. 91, and a sketch for demonstration purposes is drawn in Fig. 92. If we take two points, P and Q, on the graph (Fig. 92), and draw the ordinates PM and QN, then these ordinates represent the distances travelled in times represented by OM and ON respectively. Thus, the segment MN represents the interval of time, δt, and the distance travelled in the interval, δs, is represented by the difference between QN and PM, i.e. by QR in the figure. Thus, the *average* velocity from M to N is

$$\frac{\delta s}{\delta t} = \frac{QR}{MN} = \frac{QR}{PR}$$

since $PR = MN$. Now, in dealing with the graphs of straight lines, we called such quantities as $\frac{QR}{PR}\left(\text{or } \frac{\text{rise}}{\text{base}}\right)$ the *gradient* of the straight line PQ. In this case PQ is a chord of the curve, so we have the important fact that the *average* rate of change of a quantity is represented by the gradient of the *chord* joining the corresponding points on the graph. That is true always, and if we use y and x to denote the quantities plotted vertically and horizontally, we have

$$\frac{\delta y}{\delta x} = \textbf{gradient of chord.}$$

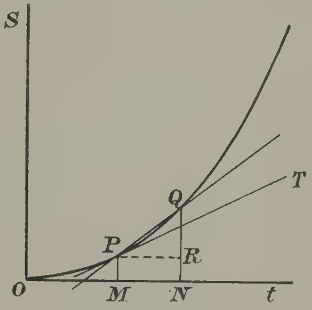


Fig. 92

We can now follow out, on the graph, the limiting process, imagining δt to become smaller and smaller, which means M approaching N, and consequently Q approaching P. Now as two points on a curve approach more and more closely, the line joining them ultimately becomes a tangent, and so we are led to the more important fact that the *instantaneous* rate of change of a quantity is given by the gradient of the *tangent* at the corresponding point of the graph. Thus, the value of $\frac{ds}{dt}$ at P in Fig. 92 is given by the gradient of the tangent PT, and, in general

$$\frac{dy}{dx} = \textbf{gradient of tangent.}$$

In Fig. 91, we have drawn the tangent to the curve at the point where $t = 2$, and taking the points A and B on it, it will be seen that the rise BC is just over 250 while the base AC is 4. The gradient is, therefore, just over $\frac{250}{4}$, agreeing very well with the result obtained in preceding lessons, 64.

Of course, it must be admitted that drawing a curve and then drawing a tangent to it is not a process that can be carried out with any very great accuracy, so that if accuracy is required, we use the formula—if we have one. With numbers derived from experiment, however, when we have no formula, and in similar cases, we have no other means of obtaining rates of change conveniently and quickly, and must perforce make the best we can of the graphical method. The value of this graphical treatment is that it enables us to visualize the process of differentiation, and also to discover some applications. Indeed, we have one application here. While drawing tangents is not an accurate method for finding rates of change, if we can find the rate of change, and thus the gradient of the tangent, we can then draw the tangent accurately, or find its equation. Thus, if we want the tangent to the curve $y = x^2 + x + 1$ at the point where $x = 1$ (and consequently $y = 3$) we first find the gradient, which is the "m" of the line, as the value of $\frac{dy}{dx}$ when $x = 1$.

Now if
$$y = x^2 + x + 1$$
$$\frac{dy}{dx} = 2x + 1 = 3 \text{ when } x = 1.$$

Thus the equation of the tangent ($y = mx + c$) must be $y = 3x + c$.
But we know that the tangent must pass through the point (1, 3) so that, putting in these values,
$$3 = 3 \times 1 + c \text{ giving } c = 0.$$
Thus the tangent is $y = 3x$. Curve and tangent are drawn in Fig. 93.

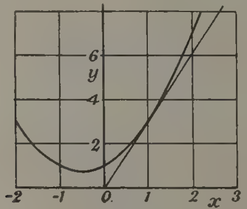


FIG. 93

Maxima and Minima. There are many instances in engineering (and elsewhere) where we seek the greatest or least value of some quantity we are interested in. If we draw a graph of the quantity in question, we can easily pick out the maximum or minimum values, as at A and B in Fig. 94. It will be seen that both at A and B the tangent to the curve is horizontal, or, in other words, its gradient is zero. This means that at A and B $\frac{dy}{dx}$ is zero. We thus see that calculus will enable us to discover these maximum and minimum values, by solving the equation $\frac{dy}{dx} = 0$. It will be noticed, however, that we have the *same* condition for a maximum as for a minimum, so we need some means of distinguishing between them. In a practical example, common sense is usually enough; we shall know whether we are seeking a maximum or a minimum. There is also a possibility of the curve behaving as at C, which, though neither a maximum nor a minimum, has a horizontal tangent. On the left of A, $\frac{dy}{dx}$ is positive, at A, $\frac{dy}{dx}$ is zero, while on the right of A, $\frac{dy}{dx}$ is negative. Thus, in the neighbourhood of A, $\frac{dy}{dx}$ is decreasing. This means that the rate of change of $\frac{dy}{dx}$ is negative. But the rate of change of $\frac{dy}{dx}$ is $\frac{d^2y}{dx^2}$, so that at a maximum, we must have $\frac{d^2y}{dx^2}$ negative. In the same way it will be seen that in the neighbourhood of B, $\frac{d^2y}{dx^2}$ is positive. At C,

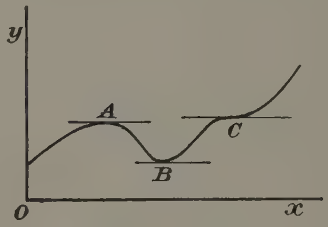


FIG. 94

on the other hand, $\frac{d^2y}{dx^2}$ is zero. Summing up

When y is a maximum or a minimum, $\frac{dy}{dx} = 0$.

At a maximum $\frac{d^2y}{dx^2}$ is positive.

At a minimum $\frac{d^2y}{dx^2}$ is negative.

We proceed to give some examples of the use of differentiation in finding maxima and minima, to illustrate the application of these rules.

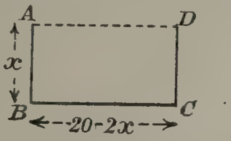


FIG. 95

EXAMPLE 1. A strip of metal, 20 in. wide, is to be bent to form the bottom and sides of a rectangular channel. Find the dimensions of the channel if its cross-sectional area is to be the greatest possible.

ABCD, in Fig. 95, represents the channel section. It will be evident that a wide channel will not be very deep and will have only a small sectional area, while a deep, and consequently narrow, channel will again only have a small sectional area. It is then a *maximum* we seek in this case. Let x denote the depth of the channel, and then we shall have $20 - 2x$ left for the bottom; the area of the section will then be given by $x(20 - 2x) = 20x - 2x^2$ sq. in.

Thus, $A = 20x - 2x^2$ is to be maximum.

Therefore, $\frac{dA}{dx} = 20 - 4x = 0$ (by the rule)

so $4x = 20$, and $x = 5$.

That is, the depth must be 5 in. and so the breadth will be 10 in. To test that it *is* a maximum, we differentiate again, and find

$$\frac{d^2A}{dx^2} = -4, \text{ which is negative}$$

so that A is a maximum.

EXAMPLE 2. A tank on a square base, open at the top, is to contain 60 cub. ft. Find the dimensions so that the least area of sheet metal may be used in its construction.

Here it is evident that a shallow tank, or a very deep, narrow one, will use more material than is used by one of more even dimensions—that we can get a *minimum* value for the area of metal in the base and sides. Calling the side of the base x, and the depth h, we have from the volume condition,

$$\text{Volume} = x^2h = 60 \quad . \quad . \quad . \quad . \quad (1)$$

also area = bottom + four sides

$$A = x^2 + 4xh \quad . \quad . \quad . \quad . \quad (2)$$

We cannot differentiate this formula for A as it stands, however, for it contains *two* quantities which vary, x and h. We can use (1) to express either in terms of the other, and it is simpler to express h in terms of x. We have

$h = \frac{60}{x^2}$ and using this in (2)

$$A = x^2 + 4x\frac{60}{x^2} = x^2 + 240x^{-1}$$

upon cancelling and expressing the fraction by a negative index, so as to be able to use our differentiation rule. Differentiating, we get

$$\frac{dA}{dx} = 2x - 240x^{-2} = 2x - \frac{240}{x^2}$$

which is to be zero if A is a minimum. Solving this, we have

$$x^3 = 120, \text{ so } x = \sqrt[3]{120} = 4{\cdot}933 \text{ ft.}$$

For the depth, $h = \frac{60}{x^2} = \frac{60}{(4{\cdot}933)^2} = 2{\cdot}467$ ft.

To test that we have a minimum, find $\frac{d^2A}{dx^2}$. It is $2 + 480x^{-3}$ which is evidently positive for the value of x we have found.

EXAMPLE 3. The loss of head in a pipe is proportional to the square of the speed of flow.

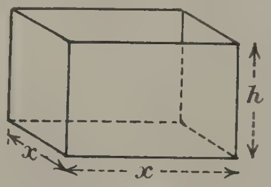


FIG. 96

Show that the issuing water has greatest energy when this loss is one-third of the available head.

Here, if h denotes the actual head, the loss at a speed v will be represented by kv^2, where k

is some constant. The net head is thus $(h - kv^2)$. The quantity of water issuing is proportional to the speed, v, so that the energy of the jet, which is the same as the net work done, will be proportional to the speed times the net head, i.e. to $v(h - kv^2)$, or $hv - kv^3$. Calling this E, E will be a maximum when $\frac{dE}{dv} = 0$.

$$\text{Now} \quad \frac{dE}{dv} = h - 3kv^2.$$

This is zero when $h = 3kv^2$, or $kv^2 = \frac{1}{3}h$. But kv^2 is the lost head, which is thus seen to be one-third of the available head h, for maximum energy. (It is *maximum*, for we must have small speed, small energy, and great speed, great loss.)

It will be clear that in (1) and (2), having found the dimensions, we can calculate the areas, while in (3), if we know the values of h and k, we can calculate v, and then, knowing also the cross-section of the pipe, we can calculate the energy of the jet.

EXERCISE No. 36

1. The speed of a piston at various angles of the crank is given by—

θ	0	30°	60°	90°	120°	150°	180°
v	0	6·44	10·1	10	7·22	3·56	0

If v is in ft./sec., and the speed is 120 r.p.m., find by drawing tangents to the graph, the accelerations at these instants.

2. Find the equation to the tangent to the curve $y = 4x - \frac{1}{4}x^3$ when $x = 2$. If the scales are equal, what angle does this tangent make with the x axis?

3. The mean effective pressure of an engine falls off at high speeds according to the law $p = 100 - \frac{v}{10}$ lb./sq. in. at a speed of v ft. per minute. Find the speed at which the rate of working is a maximum, and if the piston area is 15 sq. in., find the maximum horse-power.

4. A cylindrical can, without a top, is to hold 35 cub. in. Find the dimensions if the least area of sheet metal is used in its construction.

5. A rectangular channel is to have an area of 120 sq. ft., in section. Find its dimensions if the wetted perimeter is to be a minimum.

6. The bending moment M, at a point x ft. from the end of a beam, is given by $M = 200x - 2x^3$. Find the position and amount of the maximum bending moment.

7. In dealing with the flow of a gas (or steam) through a nozzle, we need the values of r that make $r^{\frac{2}{n}} - r^{\frac{n+1}{n}}$ a maximum. Find them when $n = 1{\cdot}4$ and when $n = 1{\cdot}3$. Also show that the general formula is $\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$.

8. The air resistance to an aeroplane is given by $R = 200\left(i + \frac{1}{125i}\right)$ lb., i being the "angle of attack" in radians. Find the value of i for minimum resistance, and the resistance at that angle.

9. A strip 39 in. wide is to be bent to form a trapezoidal channel, the sides sloping at 60° to the horizontal. Find the dimensions for the maximum sectional area of the channel.

ANSWERS TO EXERCISE No. 35

1. (*a*) $30x^5 - 9x^2 + 2.$

   (*b*) $-\frac{6}{x^3} + \frac{6}{x^4} - \frac{1}{x^2}.$

   (*c*) $1{\cdot}5\sqrt{x} - \frac{2{\cdot}5}{\sqrt{x}}$

   (*d*) $7{\cdot}2x^{0{\cdot}2} - 1{\cdot}2x^{-0{\cdot}8} - 3{\cdot}2x^{-1{\cdot}4}.$

4. 7·824.

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XXII

Estimating Weights. Proceeding with the estimate commenced in the last lesson, we have to supplement the tender drawing with a weight of the propeller. The foundation of the price-fixing in an estimate of a machine part is, of course, a rough outline drawing of the part, as given in Fig. 5 of the last lesson, together with a finished weight of the part. The term *finished weight* means the weight of the part finished machined, as distinct from the term *rough weight*, which refers to the weight of the part before machining.

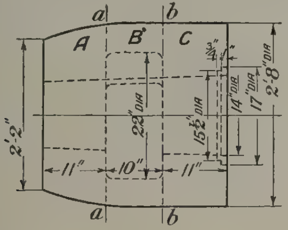


Fig. 1

Method of Dissection. Machine details, although often of irregular form, generally lend themselves to the process of an imaginary dissection into a number of solids of more regular outline. In many cases the part is so obviously built of a number of prisms that the splitting up for purposes of weight calculating is self-evident. In the example under review the form is less simple and demands special treatment.

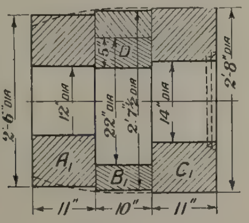


Fig. 2

We may commence our imaginary process of dissection by separating the boss from the blades and proceed to find the volume of this part of the propeller. We now have (Fig. 1) a hollow cylinder of varying internal and external diameters, and a final cutting-up of this part of the solid may be made at the planes *aa* and *bb*. The three hollow cylinders *A*, *B*, and *C*, thus produced, are not sufficiently simple for a direct arithmetical treatment, and to facilitate the calculation we may substitute three cylinders A_1, B_1, and C_1, of approximately similar volume to the cylinders *A*, *B*, and *C*, as in Fig. 2.

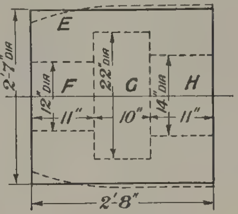


Fig. 3

Taking the cylinder A_1, the volume will be equal to the cross-sectional area of the cylinder multiplied by the length, i.e.

$$\text{Volume} = \left(\frac{\pi.30^2}{4} - \frac{\pi.12^2}{4}\right) \times 11 \text{ cub. in.}$$

Use of Table of Areas. The greater part of the above calculation may be dispensed with if a table of areas is used when we may write—

Volume of cylinder A_1
= (706·86 − 113·1) × 11
= 593·76 × 11 = 6,531 cub. in.
Volume of cylinder B_1
= (779·31 − 389·13) × 10
= 399·18 × 10 = 3,992 ,,
Volume of cylinder C_1
= (804·25 − 153.94) × 11
= 650·31 × 11 = 7,153 ,,
Volume of part D_1
= 8 × 5 × 10 = 400 ,,

Volume of boss without keyway = 18,076 ,,
,, ,, keyway = 4 × 1 × 32 128 ,,

,, ,, propeller boss = 17,948 ,,

Alternative Method. As an alternative to the above dissection we may first find the volume of a solid boss, and afterwards deduct the volume of three cylinders forming the hollow part of the detail. Fig. 3 shows the solid cylinder, of equivalent volume, drawn full, and the three cylinders representing the hollow portion shown dotted.

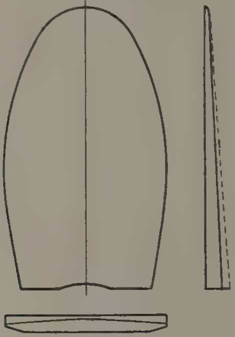


Fig. 4A

Proceeding, we have—

Volume of cylinder E

$= 754{\cdot}77 \times 32 = \underline{24{,}152}$ cub. in.

Volume of cylinder F

$= 113{\cdot}1 \times 11 = 1{,}244$,,

Volume of cylinder G

$= 380{\cdot}13 \times 10 \times \frac{45}{53}$

$= 3{,}227$,,

Volume of cylinder H

$= 153{\cdot}9 \times 11 = 1{,}693$,,

,, ,, keyway

$= 4 \times 1 \times 32. = \underline{128}$,,

,, ,, hollow portion of boss $= \underline{6{,}292}$,,

,, ,, propeller boss $= \underline{\underline{17{,}860}}$,,

The difference in the above volumes is not surprising when we consider the approximate nature of the calculations, and the methods being equally accurate our choice of method should be given to the latter which proved the speedier.

We now have to find the volume of the propeller blades and, each having the same dimensions, it will be sufficient to find the volume of one blade.

The first method which may occur to the estimator is to straighten out the blade and, assuming rectangular sections of equal area to the actual section at the boss and blade tip, thus produce an imaginary solid, as shown in Fig. 4A. Next, splitting the blade into three parts as shown in Fig. 4B, we may proceed to calculate the corresponding weights.

Volume of each section = mean thickness × width × length.

Volume of B_1 $= 3{\cdot}125 \times 34 \times 22 = 2{,}337$ cub. in.

,, ,, B_2 $= 2{\cdot}062 \times 38 \times 22 = 1{,}724$,,

,, ,, B_3 $= 1 \times 25 \times 22 = \underline{550}$,,

,, ,, one blade $\doteq \underline{\underline{4{,}611}}$,,

Alternative Method. An equally quick and more accurate method of finding the volume of a detail having such variable form as a propeller blade may be used with the aid of a planimeter. The planimeter, a common design, is shown in Fig. 5, is an instrument for recording the areas of surfaces. When using the instrument the needle E is fixed in a convenient position to allow the pointer to traverse the perimeter of the figure. Marking a starting-point F on the perimeter and placing the pointer on same, set the indicating dials D at zero. Next move the pointer over the perimeter in a clockwise direction, stopping at the point F, and read the dials D, which will record the area of figure in square inches. If the figure is drawn to a scale other than unity the true area of the figure will be equal to the area recorded divided by the square of the scale.

Applying the above method to the blade in question, we first find the area of the blade sections shown in Fig. 5 of the previous lesson by means of a planimeter. Tabulating the results we have—

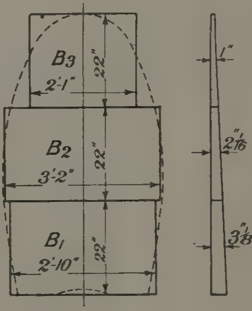


Fig. 4B

Distance of section from shaft centre $= x$	16 in.	34 in.	52 in.	70 in.
Area of drawing outline of section in sq. in. $= A$	1·92	1·59	1·16	0·45
Actual area of section in sq. in. $= 64A = Y$	122·88	101·76	74·24	28·8

In finding the actual areas Y above, it will be observed that we have divided the drawing areas A by the square of the scale to which the sections are drawn, i.e. $1\frac{1}{2}$ in. = 1 ft. Thus,

$$Y = \frac{A}{\left(\frac{1}{8}\right)^2} = 64A.$$

Proceeding, draw two lines OX, OY, at right angles, as in Fig. 6, adding lines parallel to OY at a distance from O equal to the distance of each blade section from the shaft centre. Choosing a suitable scale we may represent the area of each section, as an ordinate on the axis OX, at the corresponding distance X from O. The resulting curve Y_1, X, produces a figure X_1, Y_1, X, the area of which will represent the volume of the blade. Finding the area of the resultant figure by means of a planimeter we can correct for scale and so arrive at a value for the volume of each blade.

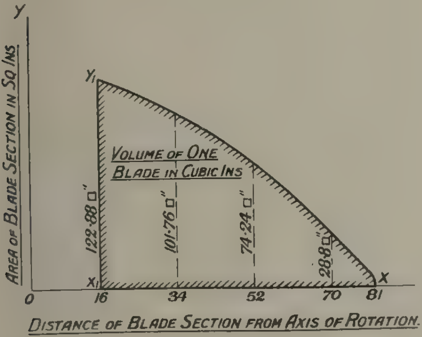


FIG. 6

Where, as in Fig. 6, different scales are used for ordinate and abscissae, the actual area is obtained by dividing the drawing area (as recorded by the planimeter) by the product of the two scales.

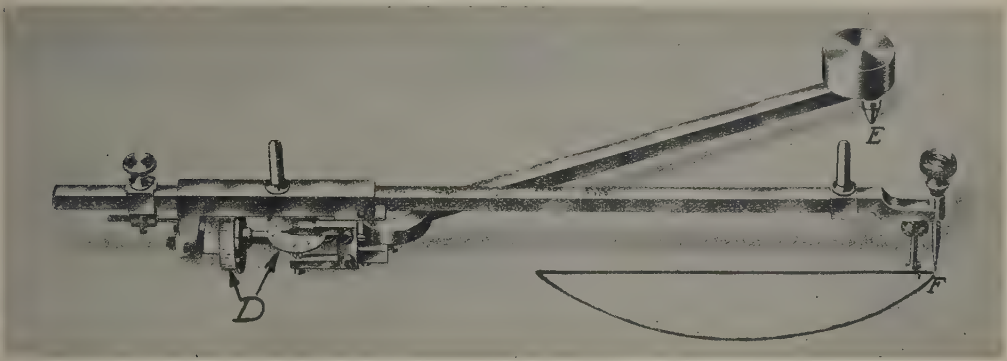


FIG. 5. A PLANIMETER

The area of the figure X_1, Y_1, X, is found to be 24·6 sq. in., from which value we find the actual area to be equal to

$$\frac{24{\cdot}6}{\frac{1}{24} \times \frac{1}{8}} = 24{\cdot}6 \times 24 \times 8 = 4{,}742 \text{ sq. in.}$$

The ordinates forming the above figure were the values A representing the areas of blade section and, consequently, each square inch of the figure represents a volume of 1 cub. in.

The volume of blade, as obtained by the alternative method, is equal to 4,742 cub. in.

Where a planimeter is available the alternative method is to be preferred and, in making a total of the volumes obtained, we may assume the alternative method has been followed for both the boss and blades.

We have

Volume of boss	-	17,860 cub. in.
,, ,, four blades	-	18,968 ,,
,, ,, propeller	=	36,828 ,,

In the above calculations we have neglected fillets, and assumed that the casting will comply with the drawing dimensions. To allow for the above omission and assumption it is advisable to increase the net calculated weight by about 5 per cent.

Taking the weight of manganese bronze to be ·32 lb. per cub. in., we have the finished weight of the propeller

$$\begin{aligned} &= 36{,}828 \times 1{\cdot}05 \times {\cdot}32 = 12{,}376 \text{ lb.} \\ &= 5{\cdot}525 \text{ ton.} \end{aligned}$$

Rough Checking. It is advisable, where possible, to check the calculation for weight on a comparative basis. This may be quickly done if weights of larger and smaller machine details of a similar form to the detail under estimate are available.

If such actual or calculated weights are available for propellers we may find the following relationship between the weight and diameter of ships' propellers.

$$W = \frac{D^3}{K}$$

where

D = diameter of propeller in feet.
W = weight (finished machined) of propeller in tons.
K = constant which for manganese bronze propellers having four blades is usually about 450.

Then for the above propeller,

$$W = \frac{13{\cdot}5^3}{450} = 5{\cdot}47 \text{ tons.}$$

The calculated weight is sufficiently near to the weight obtained by the above rough check to eliminate the possibility of a large error, such as might result from the misplacement of a decimal point.

We may now, with some confidence, submit the following estimated weight of the propeller to those responsible for fixing the estimated cost and, in turn, the quotation price.

	tons	cwt.	qr.	lb.
Finished weight of propeller =	5	10	2	0.

EXERCISE

Proceed with the exercise suggested in the last lesson, and estimate the finished machined weight of the propeller.

DIESEL ENGINES

CONSTRUCTION, OPERATION, AND MAINTENANCE

By A. Orton, A.M.I.Mech.E.

LESSON X

MECHANICAL DETAILS OF SOLID INJECTION GEAR

The principal special parts for solid injection are the fuel pump, the injection valve and the fuel piping connecting them.

There are two main systems of solid injection in common use, which have already been referred to as "continuous pressure injection" and "pump injection" respectively. In the former there is a fuel pump, fixed on and driven by the engine, or independently driven, which delivers oil into, and maintains a steady pressure in, a distribution pipe system from which a separate pipe leads to each fuel injection valve on the engine. The pump delivers a constant quantity much in excess of requirements, the surplus being let off by means of a bypass relief valve set to lift at a predetermined pressure. The injection valve is mechanically lifted, usually by means of a cam and lever, exactly as in a standard air injection arrangement, and the quantity that is admitted to the cylinder is regulated for the varying loads by varying the lift and length of the opening period of the valve. The time of opening of the valve is, therefore, a purely mechanical matter, and is unaffected by variations of pressure in the oil system. The injection valve itself is held on its seat by a strong spring so designed as to close the valve against the working pressure of the fuel oil when the lever disengages.

Such a system is illustrated diagrammatically in Fig. 38, which requires no detailed explanation. In the arrangement here illustrated the control of the quantity of fuel entering the cylinder is by means of a governor-operated wedge, which, by being moved in or out between the adjoining ends of the push rod, varies the lift of the valve. As an alternative to this arrangement the fulcrum *A* of the lever engaging with the valve spindle can be mounted on an eccentric shaft which is turned through a small angle by the governor, thus again varying the height of the lever and consequently the lift of the valve. In this case, of course, no wedge is used.

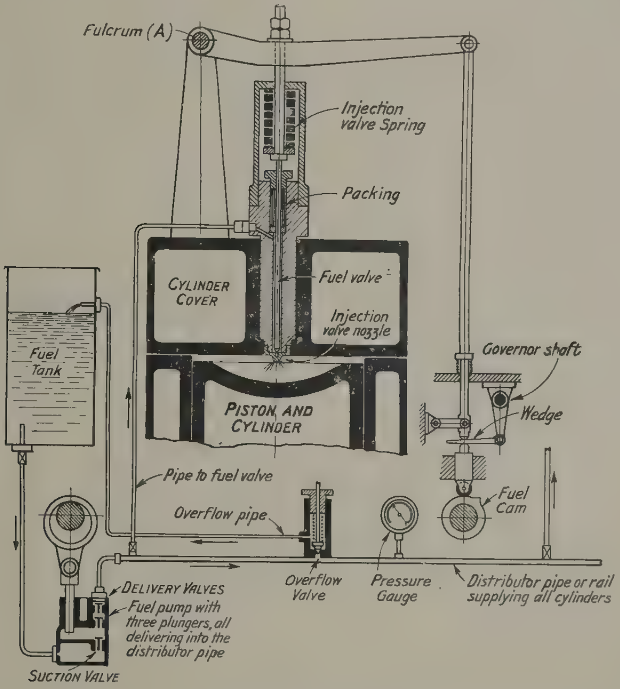


FIG. 38. DIAGRAMMATIC ARRANGEMENT OF CONTINUOUS PRESSURE SOLID INJECTION SYSTEM

In yet another arrangement the cam is made of a different shape at different axial positions (being sometimes described as a " scroll cam "), so that if it is moved axially the lift of the cam and the period of opening is varied. This design gives ideal control since the time of the beginning of injection can be maintained constant, while the cut-off at the end of injection is varied. In the previously mentioned systems, constant timing at the beginning is not obtained, and the injection tends to be either too early at high loads (with larger lift of valve) or too late at low load (with lower lift of valve). The scroll cam system, however, is better suited to hand control as required on a marine propulsion engine than for governor control as on a land engine. A simple form of injection valve for the above continuous pressure system is shown in Fig. 39.

In the second system, which is shown by the diagrammatic sketch, Fig. 40, the pump is operated by a cam so timed with the crankshaft that the pump delivers oil through the injection valve at the right moment, just before top centre of the piston stroke, and so shaped as to give the correct rate of injection into the cylinder. The injection valve is of the automatic type, controlled by a spring so designed as to permit the valve to lift as soon as a predetermined pressure in the valve casing has been reached, and to close the valve promptly as soon as the fuel supply ceases. Control of fuel quantity is by the sudden opening of a small valve in the pump chamber, sometimes called the " spill "

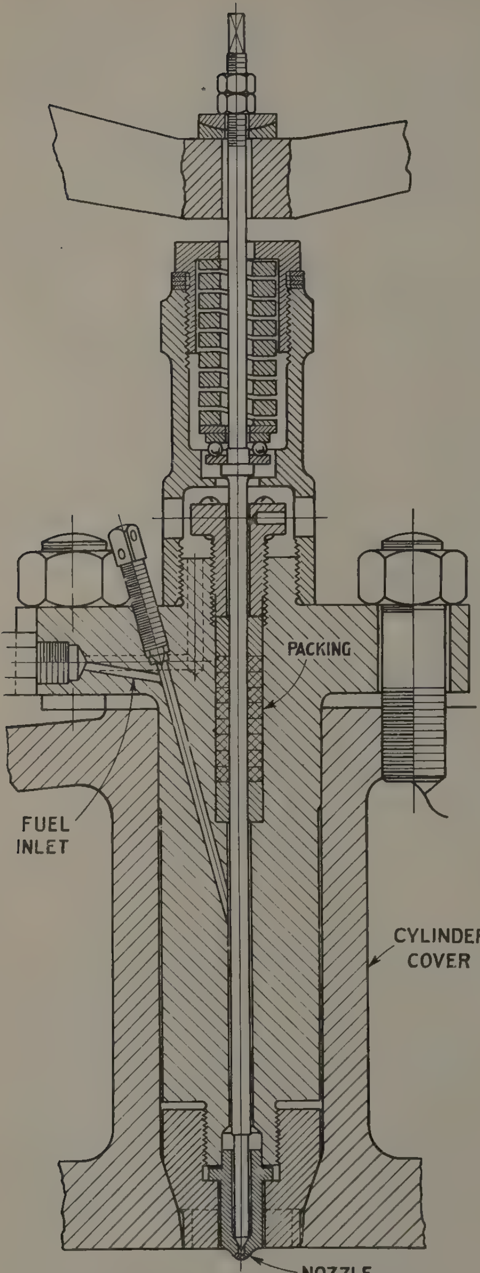


Fig. 39. Simple Form of Injection Valve for Continuous Pressure System

or " by-pass " valve, the time of opening being varied by the governor according to the load. The diagram shows a lever, one end of which is fulcrumed on an eccentric shaft controlled by the governor, and the other end is moved up and down by the plunger guide. As the guide moves upwards on the delivery stroke the tappet on the lever makes contact with the end of the " spill " or pressure breakdown valve, lifting it off its seat, and thus suddenly letting down the pressure in the pump and the delivery pipe to zero, and causing the injection valve to shut. The action of the governor is to raise or lower the fulcrum end of the lever, and thus to make the time of the spill action earlier or later in the delivery stroke, according as the load on the engine is smaller or greater. This pump system gives a constant timing of the beginning of injection and a variable cut-off, which is the ideal form of fuel control for a Diesel engine.

In both these systems the efficiency of the pulverization and distribution of the fuel is chiefly dependent on the pressure of the oil and the size, number, and disposition of the very fine holes in the injection valve nozzle. The holes vary according to the design and size of the engine, from as little as ·008 in. to about ·020 in. diameter, while pressures from 4,000 to 10,000 lb. per sq. in. are in use. The sharpness and clearness of the cut-off is also a vital factor, particularly affecting the clearness or otherwise of the exhaust.

The two diagrams given refer to no particular design, and are meant to illustrate the main principles employed. There are, of course, very many elaborations and variations in details into which it is impossible to enter here. Developments and new designs are constantly appearing, and it is yet a very open question whether, on the whole, the solid injection system will in practice show the qualities of continuous efficiency over long periods of service that has come to be expected, and is regularly obtained, with the air injection system. Probably each system will continue to find favour in special types of engines where its adoption appears more favourable, either for reasons of economy in first cost or greater reliability and efficiency in service.

LUBRICATION AND COOLING

We have now dealt very briefly indeed with the mechanical parts which may be looked upon as peculiar to the two main types of Diesel engines. We have not referred in detail to methods of lubrication or cooling, both matters of extreme importance, because these methods are not in any way different from those employed in other internal combustion engines of similar size and similar general mechanical construction. In regard to both these matters, the position is that the usual methods have been elaborated or improved in detail as the sizes and powers of Diesel engines were increased, and it might here be mentioned that for large powers, such as 500 b.h.p. and upwards, no other type of oil engine comes seriously into account.

As regards lubrication we shall merely state that, apart from the main cylinder and compressor pistons, forced lubrication is adopted for the main working parts in all engines, except a few of the open type where gravity or ring lubrication may be employed. The piston requires special treatment, and almost invariably an independent system of mechanically operated sight-feed lubrication is adopted, which enables very exact regulation and control to be attained, and positively indicates whether each and every point of application is receiving its due and necessary supply of oil. This sight-feed system has come to be regarded as essential for the proper working of all engines, except perhaps those of very small size, and is also applied to the compressor cylinders where exact and controlled supply is, if anything, even more essential than in the case of the main combustion cylinders.

Similarly, the water cooling of Diesel engine cylinders, pistons, and other working parts subject to the heat of combustion follows conventional lines, except that, as the size of the engine increases, more elaborate and refined methods have to be employed. For small engines the common thermo-syphon system as used for small gas and oil engines is quite satisfactory, while for medium sized engines a pump or other positive means of water circulation is preferred and, for large engines, is essential. Elaborations of the internal arrangements are made in large engines to ensure rapid and intense circulation at local points exposed to the full heat of the combustion gases, such as the upper end of the cylinder liner and the bottom surface of the cylinder cover. In large engines a direct water or oil circulation system becomes necessary for cooling the piston, and various satisfactory arrangements have been developed by the different makers, the cooling fluid being carried into and away from the moving piston, either by means of swinging links attached to the piston or crosshead, or by vertical trombone pipes reciprocating with the piston and passing through packings into supply and discharge chambers respectively, situated conveniently below in the crank chamber. Where oil is used as the cooling medium this is sometimes taken from the main lubricating system but, generally speaking, the supply of oil is independent. Oil has the disadvantage that it involves more or less elaborate apparatus for cooling it after it has passed through the pistons, whereas with water cooling the hot water passes away with that from the cylinder jackets and is either run to waste or cooled along with it. As a heat conductor oil is also very inferior to water.

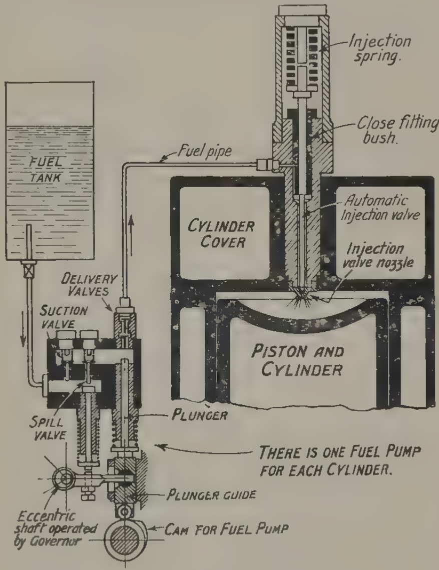


FIG. 40. DIAGRAMMATIC ARRANGEMENT OF PUMP INJECTION SYSTEM

A good idea of the design of the pistons for internal cooling and the mechanical arrangements for introducing the water can be obtained from the sectional views in Figs. 33, 34, and 35. In Fig. 34 the trombone pipe system is shown, whilst in Fig. 35 the water passes through swinging links into the crosshead, and then up the centre of the piston-rod to the hollow piston, returning by the reverse path and out through another set of swinging links.

GAS ENGINES AND PRODUCERS

By Arnold Rimmer, B.Eng.

LESSON X

PIPING AND CONNECTIONS

Gas and Air Supply. Since the relative proportions of gas and air depend on the general quality of the former, the size and type of the gas regulating valve supplied with an engine will be fixed accordingly by the makers. As stated previously, this valve is generally in the form of a conical plug, but on large producer gas engines, Messrs. Crossley, Bros., Ltd., fit a screw-down valve (of mushroom type), the arrangement of which, together with the air regulator, is shown in Fig. 59.

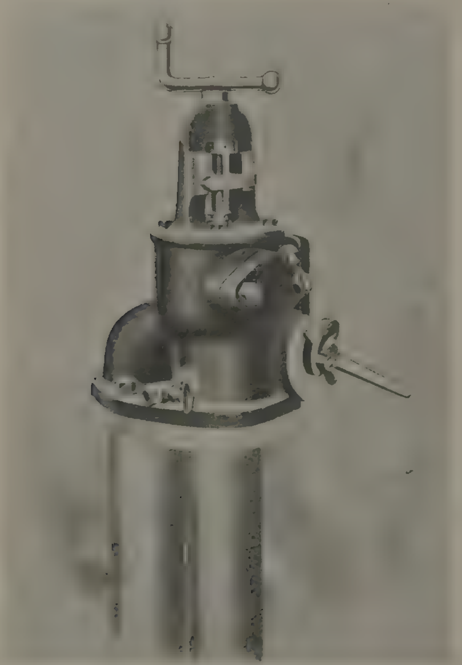


Fig. 59. Gas Valve and Air Regulator (Crossley)

If an engine is required to run either on town's gas or producer gas, two separate valves and lines of piping should be fitted, in order that the mixture may be always proportioned to suit the compression pressure.

All connecting pipes should be of adequate proportions and, in the case of producer gas, allowance should be made for the deposit of tar, etc., which gradually forms on the inside, particularly at bends, etc. The size of these gas and air pipes may be based on an average velocity in the same of 80 f.p.s., and a mean piston speed of 600 f.p.m. An approximate rule for the pipe connected to the engine is

Bore (inches) $= 0{\cdot}03 \times$ b. h. p. $+ 0{\cdot}75$.

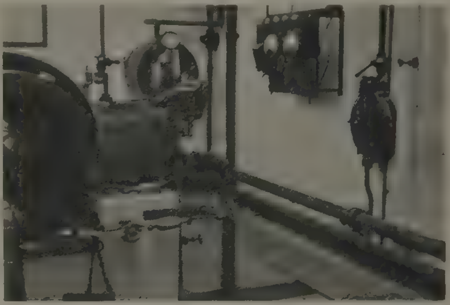


Fig. 60. Anti-fluctuator or Gas Bag (Crossley)

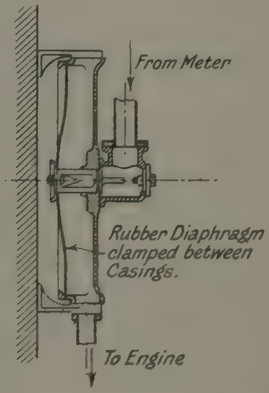


Fig. 61. Anti-fluctuator or Gas Bag (Crossley)

To prevent, as far as possible, variations in the mixture on successive induction strokes, there should be a suitable reserve of gas near to the engine. In the case of engines running on town's gas this is usually provided by an "anti-fluctuator," or "gas bag," as illustrated in Fig. 60, a section of the same being shown in Fig. 61.

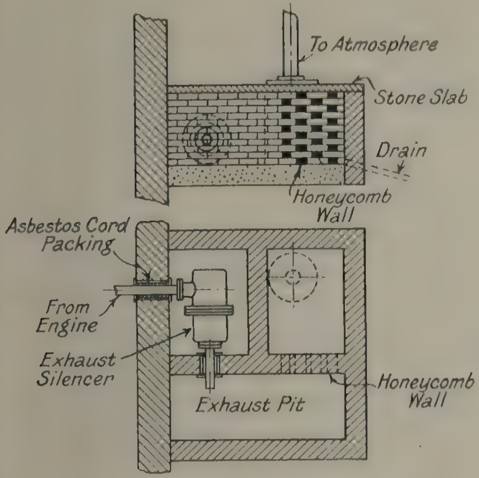


Fig. 62. Exhaust Silencing (Crossley)

It consists of a metal casing fitted on one side with a rubber diaphragm. To the centre of the latter is connected a spindle carrying a valve at one end. So long as the supply exceeds the demand this valve tends to close, but the gas already in the bag is available for sudden demands, and the consequent depression of the diaphragm causes the valve to open and the supply is replenished. These gas bags are usually erected with the rubber diaphragm next to the wall, but sufficient space should be left for the insertion of the hand between the two in order to test the working and, if necessary, get rid of an accumulation of poor gas before starting.

The size of meter for town's gas engines may be determined from the following rule—

Size of meter (in "lights") = 3·0 × b. h. p.

The pipe between the meter and gas bag is generally made slightly larger than that leading from the gas bag to the engine.

Exhaust Connections. The exhaust from the engine should be as "free" as possible, and the size of the pipe is mainly decided by experience. An approximate rule is

Bore in inches = (0·06 × b.h.p.) for large engines.
„ „ = 0·06 × (b.h.p. + 1) for smaller engines.

All unnecessary turns and bends should be avoided, and those that are essential should be of large radius. Consideration must also be given to the expansion of the exhaust pipes when the engine is running, and suitable provision made for the same.

Reference has already been made to the necessity of fitting some form of "silencer." The latter usually consists of one or more cast-iron boxes with suitable inlet and outlet connections, and having a volume four or five times that of the cylinder. The consequent expansion of the exhaust gases reduces their velocity, and in addition, the flow may be broken up by baffles, gravel, etc., contained in the silencer. The latter should not be placed too close to the engine as this will increase the back pressure, preventing the free discharge of the gases from the cylinder and lessening the consequent scavenging effect. The best position is at the far end of the engine exhaust pipe, preferably outside the engine room, with a slightly larger outlet pipe discharging into the atmosphere at roof level. On larger engines, or where extra silencing is required, a brickwork pit may be used, as shown in Fig. 62, which is a method suggested by Messrs. Crossley Bros., Ltd.

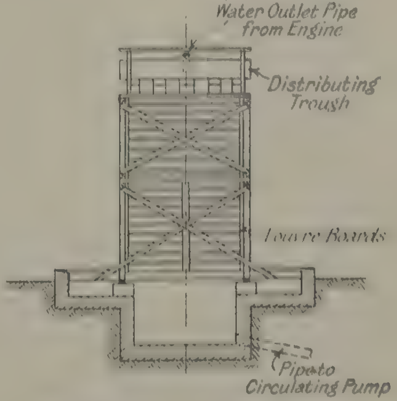


Fig. 63. Water Cooling Tower

Other makers favour the scheme of partly filling the pit with small stones.

Another point which must be borne in mind when planning exhaust connections, is the necessity of providing proper drainage facilities. The moisture present in the exhaust gases normally passes away as vapour, but when the engine is stopped this moisture condenses to a liquid which is acid in character, and is liable to corrode the metal of the pipes if allowed to remain in contact with the same.

Water-cooling System. As previously stated, it is necessary to have a water jacket on all gas engine cylinders (and in certain cases some of the moving parts have to be cooled by the same agency), and in general anything from 30–50 per cent of the heat of combustion may be thus carried away. Many considerations are involved and suitable means of adjustment are necessary if the best performance is to be obtained from any particular engine. The temperature of the water leaving the jackets should be as high as is compatible with efficient lubrication, etc. Generally, it will vary from 120 to 160° F. The jackets should have separate "open" discharge pipes, in order that the temperature may be tested and the circulation seen at a glance. The amount of water required may be taken at 30–40 lb. per b.h.p.-hr., depending chiefly on the temperatures prevailing.

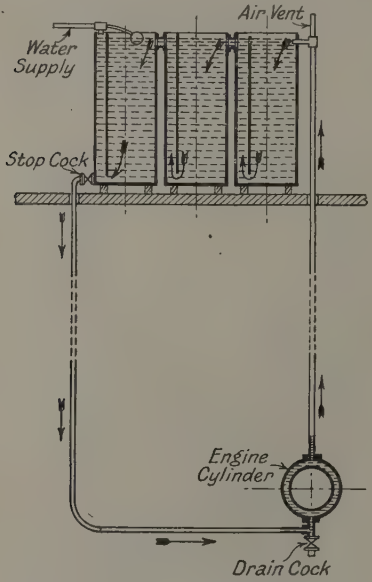


FIG. 64. WATER COOLING SYSTEM (THERMO-SYPHON)

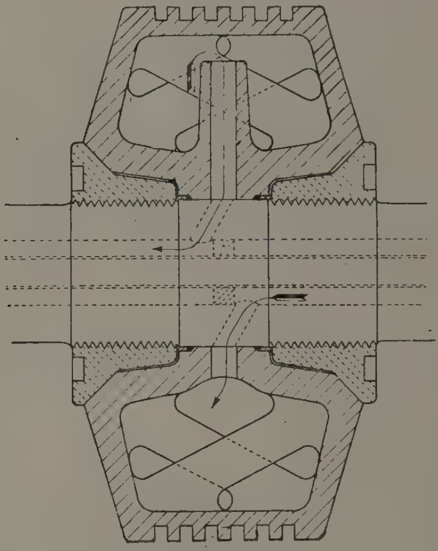


FIG. 65. WATER-COOLED PISTON (GALLOWAY)

The passages in the water jacket should be as large as possible, and designed in accordance with the varying temperatures existing at different points in the breech end and cylinder. Care must be taken to avoid pockets in which air or steam might lodge, and provision must be made for cleaning, in order that the "scale" which is usually formed may be periodically removed. Otherwise the efficiency of the cooling surface becomes impaired. If soft rain-water is used there will not be much trouble in this respect, and small quantities of soda added to ordinary water will be useful, though complete "softening" of the latter requires the use of certain chemicals, according to its composition.

Where the water supply is cheap and plentiful, the outlet water may be allowed to run to waste (though a storage tank is advisable for emergencies). Generally, however, it is arranged to use it over and over again, it being cooled each time by exposure to the atmosphere. In small and medium sized engines this may be effected by passing the water through tanks in a suitable position, while for larger engines a mechanical cooler or some form of "cooling tower" may be desirable (Fig. 63). The latter consists of many tiers of wooden planking, crossed and recrossed so as to expose the maximum amount of surface to the atmosphere. Over these a number of small streams of water descend and, while the air can reach the same from all directions, the louvre boards fitted on the outside prevent the spray being blown away by the wind. Of course, any loss due to absorption of moisture by a dry atmosphere requires to be made good.

The circulation is usually effected by the "thermo-syphon" or natural method which is illustrated in diagrammatic form in Fig. 64. This system, of course, depends on the reduction in the density of water with increasing temperature, and the greater the height of the tanks above the centre line of the engine, the greater will be the force causing the circulation. Hence it is obvious that both from the point of view of rapid circulation of water in the cylinder jacket and also adequate cooling in the tanks, the latter should be placed in the highest and most exposed position possible. Protection against frost should be provided by draining the whole system when necessary, rather than placing the tanks in a low or sheltered position, where they become comparatively inefficient. The approximate size of the tanks may be fixed by allowing a minimum storage capacity of 30 gal. per b. h. p. A good circulation is assured by making the effective height of the tanks two and a half to three times their diameter, and fitting an internal baffle plate as shown. In the case of larger engines, the above natural method of circulation is not very satisfactory, and pumps are generally installed, either in connection with tanks, cooling towers, or a stream, if available. The power absorbed by such pumps is very small.

As previously stated, if more than about 150 h.p. is developed per cylinder (corresponding diameter about 22 in.), it becomes necessary to apply water cooling to the piston. The latter, of course, is in varying and rapid motion, but the supply of water must be continuous and many ingenious arrangements have been devised for this purpose. Because of the reciprocating motion of the piston the water must be supplied at a pressure of about 50 lb. per sq. in., and great care is required in the design of the telescopic or other form of glands if the latter are to work freely without leakage. The piston of the Galloway engine is shown in Fig. 65, and a section of the engine in Fig. 66.

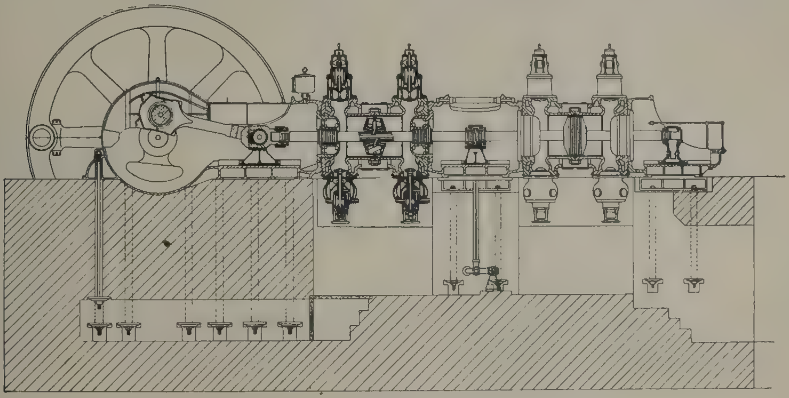


FIG. 66. GALLOWAY GAS ENGINE

Water enters and leaves the pistons via the hollow rods, and it is led to the latter by rocking pipes connected to the coupling between the pistons.

The main difficulty in the water cooling of the exhaust valve is not the supplying of sufficient water, since the actual amount required is comparatively small, but rather the providing of passages of such an area as will prevent choking up by deposit, etc. Some makers, therefore, fit a solid exhaust valve and concentrate their attention on efficient cooling of the parts immediately surrounding the same. The exhaust valve casing is usually a separate casting bolted to the main cylinder and, in some cases, the valve seating may also be a separate piece and is, therefore, easily renewable.

REFRIGERATING MACHINERY

By G. W. Daniels, M.Eng., Wh.Ex., A.M.I.Mech.E.

LESSON III

MACHINES—(*contd.*)

Multiple Effect Compression. A system which has come into use, particularly for CO_2 machines, is known as *multiple effect compression,* and is illustrated diagrammatically in Fig. 16. In this system the CO_2 from the condenser is not expanded through the regulating valve directly into the refrigerator coils, but first passes into a vessel *E* called a multiple effect receiver. This receiver is maintained at some pressure intermediate between that in the condenser and evaporator. The function of the receiver is to separate the CO_2 vapour, which is always formed during the expansion past the regulating valve, from the liquid CO_2. The latter is passed through a second automatic regulating valve *F* into the refrigerator coils, while the CO_2 vapour is returned direct to the compressor.

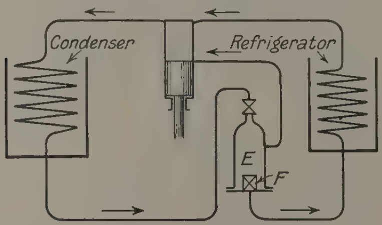


Fig. 16. Diagram of Multiple Effect Compression System

Such a system possesses advantages where the temperature of the condensing water is high and the refrigerator temperature low, because under these conditions quite a large proportion of the liquid CO_2 leaving the condenser evaporates during the passage through the regulating valve and is, therefore, useless so far as the cooling of the brine or other substance round the refrigerator coils is concerned. The expansion in two stages in the multiple effect system greatly reduces the amount of liquid CO_2 evaporated in this way and, therefore, increases the output of a given size of machine. The vapour from the receiver usually enters the compressor through ports overrun by the piston, as already described for the two pressure system of working. The low pressure gas first drawn into the compressor is therefore compressed up to the receiver pressure by the gas from the receiver, and not by the machine, and consequently some economy is obtained in the power required per unit of refrigeration.

Such a system is applicable to other refrigerants than CO_2, but it is with CO_2 that the best results are obtained.

Mechanically-operated Compressor Valves. An interesting development of recent years has been the introduction of mechanically operated compressor valves in place of the usual automatic spring controlled valves. So far, only machines using ammonia have been built on these lines, and the type of automatic valve employed is that known as the sleeve valve and hitherto employed on motor-car engines. Fig. 17 (facing page 1113) shows the operation of a sleeve-valve machine in which the sleeve acts as both a suction and delivery valve. *D* is the compressor piston, *E* the sleeve valve having two sets of ports in it, *C* the safety head, *A* the suction, and *B* the delivery pipe. From a study of the diagrams it will be seen that during the suction stroke, the lower set of ports come opposite the suction opening *A* and allow gas to enter the cylinder, the upper set of ports being meanwhile closed by the safety head. During the discharge stroke the upper set of ports are, by the movement of the sleeve, brought past the end of the safety head, and so permit the compressed gases to escape from the cylinder into the annular space around the safety head. During the discharge stroke the suction ports are, of course, closed. This type of valve is only used on comparatively small size machines, because it is impossible so to time the movement of the sleeve and piston that the ports open and close at exactly the right moments for all the varying conditions under which the refrigerating machines have to work. The timing can only be correct for one set of working conditions and, therefore, with any other conditions there are some losses and inefficiencies. In small size machines, however, such losses are said to be counterbalanced by the simplicity due to the absence of automatic valves.

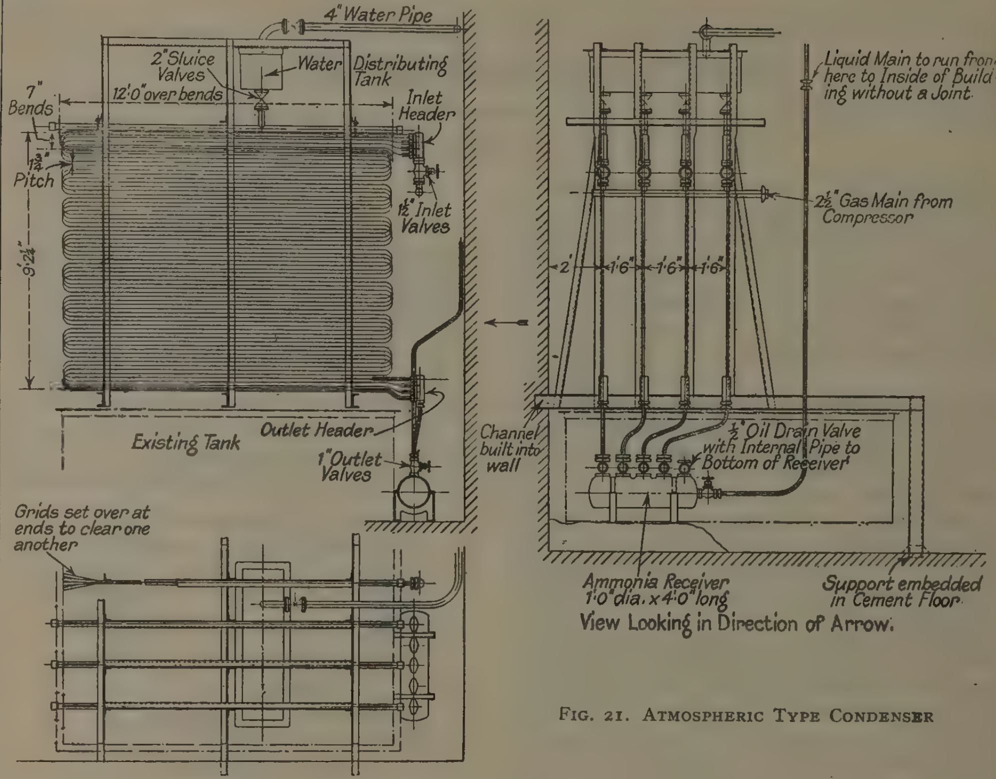

4" Water Pipe
2" Sluice Valves
Water Distributing Tank
7" Bends
12'0" over bends
Inlet Header
1¾" Pitch
1½" Inlet Valves
9'2¼"
Existing Tank
Outlet Header
1" Outlet Valves
Grids set over at ends to clear one another
Liquid Main to run from here to Inside of Building without a Joint
2½" Gas Main from Compressor
2'
1'6"
1'6"
1'6"
Channel built into wall
½" Oil Drain Valve with Internal Pipe to Bottom of Receiver
Ammonia Receiver 1'0" dia. x 4'0" long
Support embedded in Cement Floor
View Looking in Direction of Arrow.

FIG. 21. ATMOSPHERIC TYPE CONDENSER

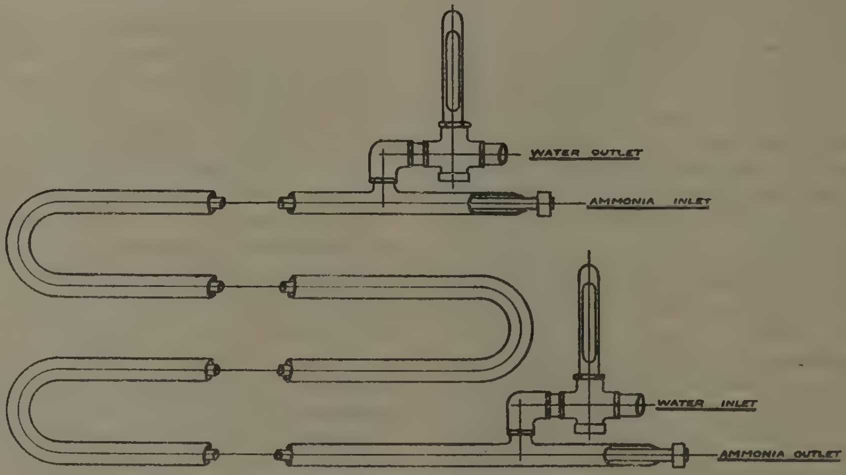


FIG. 22. SMALL DOUBLE-PIPE CONDENSER

Fig. 18 (facing page 1113) shows diagrammatically the operation of a large size sleeve-valve machine. In this the sleeve valve only operates the suction. The delivery is through an automatic spring-loaded valve in the safety head. Fig. 18 is also interesting in so much as it shows clearly the operation of a machine with two suction pressures, and the inrush of gas at the higher suction pressure will be noted.

Fig. 19 shows the external appearance of a sleeve-valve machine, while Fig. 20 is a photograph of the piston, cylinder, and sleeve valve.

All refrigerating machines are equipped with a set of fittings which have now become practically standard. First and foremost, stop valves are provided on the compressor suction and delivery to enable the compressor to be quite shut off for examination. Smaller valves and connections are also provided to enable the compressor to be by-passed when starting up. This is essential if the machine is driven by a gas or oil engine which cannot start against a load. These connections also serve to cross over the suction and delivery pipes so that, if necessary, the compressor can pump the refrigerant out of the condenser, and not into it as in ordinary running. The condenser can thus be emptied for repairs. A strainer or dirt trap is fitted on the compressor suction to prevent dirt or grit entering the compressor cylinder, and an oil trap is fitted on the discharge to remove any lubricating oil carried over by the refrigerant and keep it out of the condenser coils. In ammonia plants it is customary to fit an ammonia reservoir or "pig" at the condenser outlet. This reservoir serves as a storage vessel for the liquid ammonia. Pressure gauges are also always fitted on the compressor suction and discharge.

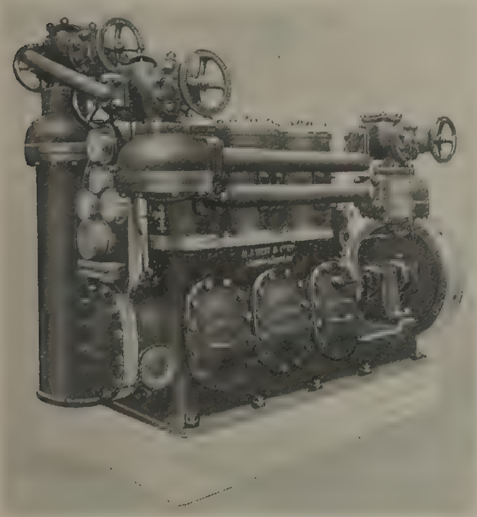


Fig. 19. Double-suction Sleeve-valve Machine
(H. J. West & Co. Ltd.)

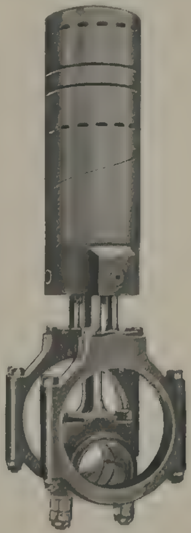


Fig. 20. Sleeve Valve for Ammonia Compressor
(H. J. West & Co. Ltd.)

CONDENSERS AND EVAPORATORS

Not the least important parts of the refrigerating plant are the coils of pipe serving as condensers and evaporators. If these parts are of insufficient size, or wrongly proportioned, the plant will not work well, however good the compressor.

The most common type of condenser for land use is what is called the atmospheric condenser. Fig. 21 illustrates this. Such condensers consist of several sheets of piping, fastened to an angle iron framework, each sheet being arranged in a vertical plane. Each of these sheets of pipes is composed of three or four continuous grids formed by electrically welding together lengths of piping, and suitably bending the pipe to form the grids. The three or four grids forming the sheet are interlaced as shown in

Fig. 21, the bends at the ends of the pipes being turned outwards slightly to the right and left as required. This construction enables a large amount of piping to be fitted in a moderate space. The end pipes of each grid are jointed to headers of cast iron or welded steel. A stop valve is usually fitted at each header to enable any one sheet of pipes to be shut off. Over the topmost pipe in each sheet is fitted a length of piping having a slot in its uppermost part, and this pipe is fed with water from an overhead tank supported on the framework. The water escapes from the slotted pipe, flows down its sides, and thence trickles down the sheets of grids forming the condenser. The water is collected in a large tank underneath the condenser and re-circulated by a pump. This type of condenser is always placed in the open air, being, if necessary, protected by wooden louvre boards from the sun and strong winds, which would otherwise blow the water off the pipes. The condenser really serves two purposes; it condenses the refrigerant and, at the same time, acts as a cooling tower for the condensing water, because a small portion of this evaporates into the atmosphere as the water trickles down the pipes, and this evaporation cools the remainder of the water sufficiently to enable it to be used over again. Hence its name "atmospheric condenser." Such condensers are very economical of water but require a large amount of pipe surface.

If water is plentiful, as happens when the plant is situated on a river or canal bank, then a submerged condenser may be used. This consists simply of coils placed in a tank with water circulated over them. The water is used only once and then goes to waste. Such condensers do not require much pipe surface, but can only be used where water is plentiful.

Another type of condenser used especially for small plants is the double-pipe condenser. This consists essentially of a series of concentric pipes, one inside the other. The refrigerant usually is condensed inside the smaller pipe while the cooling water is passed through the space between the two pipes. Many different constructions of double-pipe apparatus are possible.

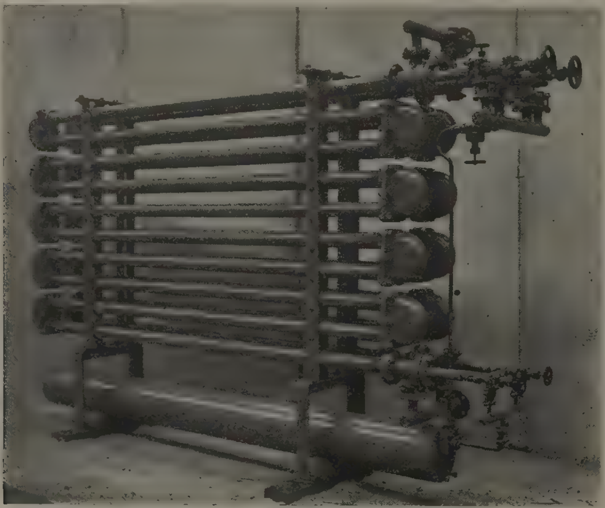


Fig. 23. Double-pipe Condenser

Fig. 22 shows the simplest, used for the smallest plants. In this the two pipes are placed inside one another and bent to give the form shown.

Fig. 23 shows a more elaborate type used for larger installations.

Condenser Surfaces. The following rules indicate approximately the amount of pipe surface and quantity of cooling water required for different types of condenser, all being given per ton refrigeration—

Atmospheric type .	35 to 40 sq. ft. surface. 230 gals. water per hour.
Submerged type .	25 to 30 sq. ft. surface. 60 to 65 gals. water per hour.
Double-pipe type .	10 to 22 sq. ft. surface. 120 gals. water per hour.

Condenser coils may take several forms: flat grids, as in Fig. 21; circular coils, as in Fig. 24 (*a*); "oval coils," as in Fig. 24 (*b*); or even rectangular coils, as in Fig. 24 (*c*). When one or more coils are placed one inside the other, they are described as being "nested."

Coils for ammonia are always made of iron or steel, but for CO_2, copper is often employed, especially if the condensing water is salt or likely to cause iron to corrode. Copper coils are expensive but last a very long time.

The refrigerator or evaporator coils in which the refrigerant evaporates and produces the cold, often take very similar forms to those just illustrated for condensers, the exact arrangement depending upon whether we want to cool a liquid or a gas. As we shall refer to these coils again when we treat of applications of refrigeration to definite purposes we need not consider them further at this time.

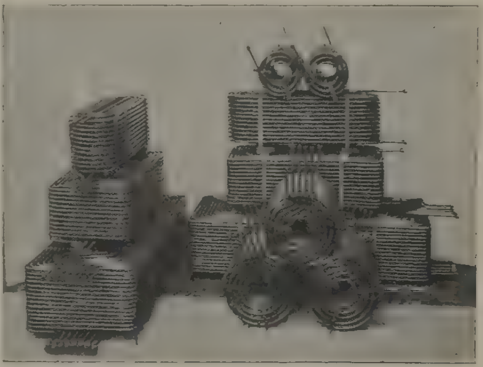


FIG. 24

SOLDERING, BRAZING, AND WELDING

By E. A. ATKINS, M.I.MECH.E., M.I.W.E.

LESSON II

THE WELDING OF METALS

THE welding together of two pieces of the same metal without the introduction of any form of solder, spelter, or other metal, is termed autogenous soldering or auto-welding; the latter term, of course, meaning the self-jointing of a single metal.

At the present time almost any kind of metal or alloy can be welded by the application of one or other of the methods of gas or blow-pipe welding, or the electric-arc, resistance or other processes to which the use of electricity is applied.

Blacksmith's Welding. This is the oldest method of auto-welding, and no doubt dates back to the time when iron was first made, as it is now known that comparatively small sized pieces of wrought iron were originally made by the welding together of several smaller lumps.

Whilst the blacksmith's weld is very common it is not by any means always an efficient method of jointing, as it is just one of those operations which bring in the personal element of the workman more than many others. No exact degree of efficiency can be fixed as to the strength of a blacksmith's weld, for whilst it is often said that the strength of a fire weld gives 80 per cent of the strength of the solid metal, actual testing shows that this is not by any means true. In some cases the strength of the weld when made by an experienced blacksmith may be equal to that of the solid bar, yet in other cases it may be anything down to as low as 30 per cent efficiency.

For good fire welding the ends of the bar should be heated to the right temperature, and it is certainly an advantage to use a flux of fine sand or a compound of equal proportions of well-heated borax and salammoniac, which will assist in removing any oxide from the surface of the metals which are to be joined. Any layer of oxide or slag which is bottled up in the weld is fatal to a strong joint, as any part of the joint which is not welded inevitably leads to a reduction of strength.

Oxy-acetylene Welding. During the last few years this method of welding has made enormous strides, and this is not to be wondered at when one considers the advantages of this particular method of jointing metals. The oxy-acetylene blow-pipe flame gives a very high temperature running up to between 6,000 and 7,000° F., thus giving the temperature necessary to melt any of the ordinary metals. In this system of welding, too, the blow-pipe flame, if properly formed by the right mixture of gases, gives certain products of combustion which act as reducing agents, and thus tend to keep the metal from being permeated with oxides or allowing scale to form on the surface when the metal is in a molten state.

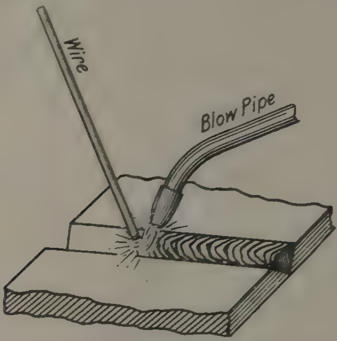


FIG. 5. OXY-ACETYLENE WELDING

Oxy-acetylene welding permits the fusing together of bars or plates of iron or steel from the thinnest sheet to almost any thickness of plate that is used in the boiler shop. With great care it is possible to make the welded portion as strong as the solid bar or plate, but generally it is the safest plan to assume that the welded part will be slightly weaker. A competent welder can certainly make joints which are equal to 95 per cent of the strength of the solid plate.

Those who commence to use the oxy-acetylene process should be warned of the absolute necessity of making a large number of trial joints in metals, and to test them by either pulling or bending to ensure the production of work which will have maximum strength and toughness.

All kinds of work should be properly prepared before any attempt is made to weld same. Steel or iron plates which are over $\frac{1}{8}$ in. in thickness should be bevelled, so that when they come together a " V " is formed, which should be about as wide at the top as the plates are in thickness.

The feeding wire should be of a quality suitable for the work and never of a greater diameter than the thickness of the work, and in the case of heavy plates should not be greater than $\frac{1}{4}$ in. diameter. The method of using the blow-pipe and applying the feed wire is shown in Fig. 5.

For " cutting " iron or steel a special form of blow-pipe is used, in which it is so arranged that a central stream of oxygen can be brought into the flame which rapidly cuts the metal away by oxidation.

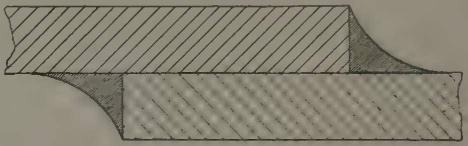


FIG. 6. ELECTRIC ARC WELDED LAP JOINT

Oxy-hydrogen Welding. The oxy-hydrogen blow-pipe has been in use in connection with lead burning and similar operations for a long time. For several reasons it cannot rank with the oxy-acetylene method for welding purposes. Apart from the fact that the temperature of the flame is lower than that of the oxy-acetylene blow-pipe, the product of combustion is a considerable drawback. In the burning of hydrogen, water vapour is always formed, which causes the formation of oxide when it comes into contact with molten or hot iron.

Oxy-coal Gas Welding. The use of oxy-coal gas welding is confined to low temperature work, such as lead burning, brazing, etc., therefore it has a much more limited use than in the case of oxy-acetylene.

Oxy-benz Welding. A considerable amount of investigation in connection with the use of light mineral oils has been carried out on the

Continent, but whilst under certain conditions these may give some particular advantages, as a rule they cannot compete with oxy-acetylene welding.

Oxy-water Gas Welding. This gas and several other gases of this order have been used with some degree of success for particular kinds of welding purposes, but up to the present, as compared with acetylene, they do not warrant any great consideration except when due to some local conditions.

Thermit Welding. Some years ago in this country, considerable interest was aroused by the application of thermit welding to the jointing of tramway rails and similar class of work, but at the present time it has nothing like the scope of acetylene and electric welding. The process is very simple. A mixture of powdered aluminium and iron oxide is placed in a crucible, and fired at the upper part. As aluminium has great affinity for oxygen it readily robs the oxide of iron of this element, combining with it to form oxide of aluminium or alumina. Meanwhile the molten iron trickles down through the heated mass and runs through a hole in the bottom of the crucible into a mould placed round the ends of the bar which required jointing.

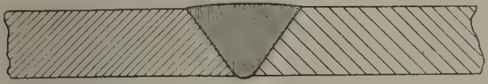


Fig. 7. Electric Arc Welded Butt Joint

Electric Welding. There are really only two ways in which steel can be welded by the use of an electric current. One is known as the "Resistance" method of welding, where the electric current is used simply as a source of heat, the heat generated being due to the resistance set up at the joint of the metal. The other is known as the "Arc" method of welding, the metal to be welded being either melted or vaporized in the electric arc.

Generally there are three kinds of resistance welding, viz., butt welding, spot welding, and contact welding.

Butt welding is the method adopted when the two ends of a rod, bar, tube, or wire are butted together and an electric current passed through the joint of sufficient strength to gradually heat up same to the melting point of the metals, these being then squeezed together and welded.

Spot welding is used to take the place of riveting in iron sheets or thin plates. The sheets or plates are squeezed together over a small circular area in a grip. The electric current is then passed through the part held and melts the portion of the sheet between the grip, bringing this up to a welding temperature and thus uniting same.

Contact welding is generally used for the making of reinforcing fabric, or other net-like structures which are made out of bars or wire. In this the crossed wires are simply pressed together, and at the same time an electric current passed through the joint which melts the points of contact of the two wires, and thus effects a welded joint over a small area.

Arc Welding. Arc welding of recent years has taken many forms, and these will be dealt with in a further article. At the present time the method in most general use is the one where a wire metallic electrode is used, the latter being coated with some form of flux. The electrode itself also forms the filling material, this being melted away as progress is made with the joint formation. In Fig. 6 an electrically welded lap joint is shown, of the type used for the shell plates of large ships. The shaded filleted parts are those filled in by the use of wire. In Fig. 7 a similarly welded butt joint is shown, this being used where the unbroken plate surface is required. The strength of the lap joint is usually greater than that of the plate. In butt welds, however, the strength of the joint rarely equals that of the solid plate.

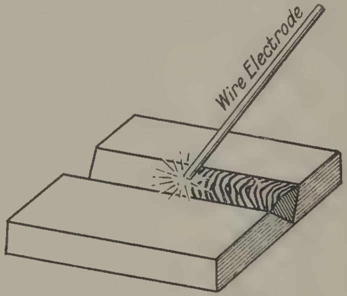


Fig. 8. Metallic Arc Welding

The method of applying the wire electrode to weld two plates together is shown in Fig. 8.

As the metallic arc method of electric welding is now so commonly used and is of such great importance to the engineer, one or two further papers will be given explaining methods of working and the kind of joints which can be formed.

APPLIED MECHANICS

By Geo. W. Bird, Wh.Ex., B.Sc., A.M.I.Mech.E., A.M.I.E.E.

LESSON XXI

ELEMENTS OF MECHANISM
(contd.)

Definition of Machine

If we connect together a number of links or elements in such a manner that *relative motion* between the parts can take place, one link being fixed, we obtain a *Mechanism*. Such an arrangement may be quite useless for the purpose of transmitting *force* in any direction and, in fact, may be regarded as the skeleton frame of a machine. When, in addition, the elements or links possess the property of rigidity to a sufficient degree, that they can transmit the forces for which they are designed without undue deformation and, further, when the movements of the several links take place in predetermined directions, then we are justified in referring to our combination of links as a Machine. The function of a machine is to receive a supply of energy in some available form and to so modify or convert it that it takes on a more convenient form for the particular purpose in view. Turbines and engines of all kinds are examples of machines, since they convert the energy in the steam, oil, or gas supplied to them into the mechanical energy of rotation of the main or crankshaft. Workshop machines, in general, also satisfy our requirements, in that they convert the mechanical energy of rotation of a driving shaft or pulley to the useful work of some machine operation.

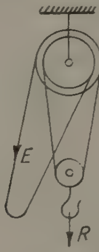


Fig. 117

Machines, however, need not be of complicated form, nor need they comprise a large number of elements; a wheel and axle with its bearings form a machine, so also do a lever and its fulcrum, and a wedge with the bodies which it separates. From the above, the reader will see that at least two elements are required and, that constrained relative motion between these two elements must be possible, before we can claim to have satisfied the requirements of a *Machine*. This, of course, excludes those combinations of rigid bodies known as structures, since no constrained relative motion takes place between the several elements.

Velocity Ratio. This is a term usually applied to a machine as a whole, and denotes the ratio between the velocity of the point of application of the effort and that of the point at which the useful work is done. If, as an example, we consider a pair of chain blocks, Fig. 117, as used for lifting heavy bodies, then the velocity ratio is measured by the fraction—

$$VR = \frac{\text{velocity of the ``fall'' } E \text{ (where the effort is applied)}}{\text{velocity of the load } R};$$

and, since the time interval during which the respective movements take place are equal, we actually measure and compare these movements

$$\text{or, } VR = \frac{\text{length of ``fall'' chain hauled at } E}{\text{distance which } R \text{ is raised}}$$

$$\text{or, } VR = \frac{\text{movement of effort}}{\text{movement of load}}$$

Mechanical Advantage. The *Mechanical Advantage* (MA) of a machine is the measure of an advantage gained by the use of the machine. Referring again to our pair of chain blocks, Fig. 117, it is clear that we gain a certain advantage when we employ this machine in raising heavy weights; the proportions of the machine are such that, by the application of quite normal efforts, we are able to raise weights many times greater than the efforts which we exert. This advantage is measured by the ratio—

$$MA = \frac{\text{resistance overcome}}{\text{effort applied}}$$

Efficiency. It is of the highest importance to an engineer that he should be quite familiar with the notion of *efficiency*; used in its widest sense, it is *efficiency* which determines which type of machine we employ for a particular purpose. In our present study, however, we must restrict our attention to that of *mechanical efficiency* and exclude all other considerations, such as convenience, reliability, initial and operating costs, etc., which, in the final count, determine the type of machine adopted.

In order that a machine may do a particular piece of work, it is necessary to expend a certain amount of energy in driving the machine and, of this energy, a part is invariably found to be consumed in overcoming the internal resistance of the machine itself. This results in there being less energy available for doing useful work than was supplied to the machine, and the ratio between the useful work done by the machine and the total energy supplied is termed the *efficiency* of the machine, hence

$$\text{efficiency} = \frac{\text{total energy supplied to machine} - \text{all losses}}{\text{total energy supplied to machine}}$$

or,

$$\text{efficiency} = \frac{\text{work done by machine}}{\text{work supplied to machine}}$$

The following worked example is given to make clear the various terms explained above, and also to show the various relations which exist between the quantities.

EXAMPLE. A machine, whose velocity ratio was found to be 20, was tested and the following data obtained—

Effort applied to machine . .	8	20	33	43	55	68	80	92	104	116
Resistance overcome by machine .	0	100	200	300	400	500	600	700	800	900

Calculate and tabulate the corresponding values of MA, friction and ideal efforts and efficiencies and plot graphs of effort (E), ideal effort (IE), MA and efficiency with resistance overcome.

By *Ideal Effort* is meant "that effort which would be just sufficient to operate an ideal machine." An ideal machine would operate at 100 per cent efficiency, and since,

$$\text{efficiency} = \frac{\text{work done by machine}}{\text{work given to machine}};$$

we may write

$$100 \text{ per cent} = \frac{\text{resistance} \times \text{movement of resistance}}{IE \times \text{movement of } IE}$$

$$\text{or } 100 \text{ per cent} = \frac{\text{resistance} \times 1}{IE \times VR}$$

$$\therefore \quad IE = \frac{\text{resistance}}{VR}$$

Again, to determine the actual efficiencies at the various loads we make use of the expression—

$$\text{efficiency} = \frac{\text{work done by machine}}{\text{work given to machine}}$$

or, efficiency

$$= \frac{\text{resistance} \times \text{movement of resistance}}{\text{effort} \times \text{movement of effort}}$$

$$\therefore \quad \text{efficiency} = \frac{MA}{VR}$$

Since the original data was obtained experimentally, we may expect to find some slight errors in the given values; these can be eliminated by plotting a graph of *Efforts* and *Resistances* and tabulating a series of corrected efforts, as under.

The *Friction Effort* (FE) is the excess of the actual effort above the IE.

RESISTANCE AND EFFORT

This graph is seen to be a straight line and hence it can be expressed in the form—

$$E = aR + b \quad . \quad . \quad . \quad (1)$$

"a" and "b" being constants.

Corrected efforts . . .	8·53	20·36	32·13	43·97	55·8	67·63	79·53	91·36	103·19	115·02
Mechanical advantage . .	0	4·91	6·224	6·82	7·17	7·39	7·54	7·66	7·75	7·82
Ideal effort	0	5	10	15	20	25	30	35	40	45
Friction effort . . .	8·53	15·36	22·13	28·97	35·8	41·63	49·53	56·36	63·19	70·02
Efficiency per cent . .	0	24·55	31·12	34·1	35·85	36·95	37·7	38·3	38·75	39·1

From the graph of E and R, we select two points; thus, when $R = 800$, $E = 103{\cdot}2$; when $R = 200$, $E = 32{\cdot}2$.

Substituting these values of E and R in equation (1) we have—

$$\left.\begin{array}{r}103{\cdot}2 = (a \times 800) + b \\ 32{\cdot}2 = (a \times 200) + b\end{array}\right\}$$

which, on solution, gives $a = \frac{71}{600}$ and $b = 8{\cdot}53$.

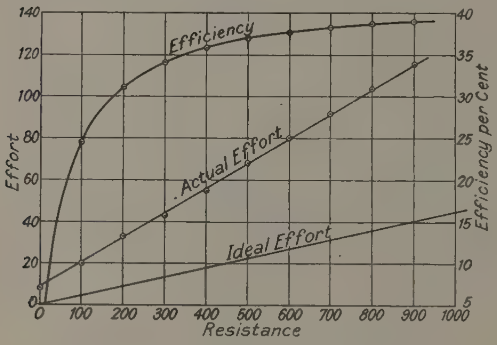


FIG. 118

The relation between effort (E) and resistance (R) can now be stated as—

$$E = \frac{71}{600}R + 8{\cdot}53 \quad . \quad . \quad (2)$$

RESISTANCE AND EFFICIENCY

Since $E = aR + b$,

multiplying through by $\frac{1}{E \times VR}$ we obtain;

$$\frac{E}{E \times VR} = \frac{aR}{E \times VR} + \frac{b}{E \times VR};$$

which reduces to

$$\frac{1}{VR} = a \times \text{efficiency} + \frac{b}{VR} \times \frac{1}{E} \quad . \quad (3)$$

which is in the form of a straight line equation, efficiency and $\frac{1}{E}$ being the two variables. The graph of these variables is shown in Fig. 119. Substitution of the known constants in (3) gives—

$$\frac{1}{20} = \frac{71}{600} \times \text{efficiency} + \frac{8{\cdot}53}{20 \times \text{effort}} \quad . \quad (4)$$

LOWER AND HIGHER PAIRING

In our earlier consideration of the essentials of a *Machine*, we spoke of a number of successfully constrained *elements* or *links* being connected together. Any two links connected together so that the motion of one is successfully constrained by the other, form a *pair* and a series of pairs form a *kinematic chain*. If we *fix* one link of our kinematic chain and thus completely determine the motion of all the remaining links, we obtain a *closed kinematic chain*. The simplest *kinematic chain* consists of four linked pairs, as shown in Fig. 120, which is termed a *Quadric Cycle Chain*.

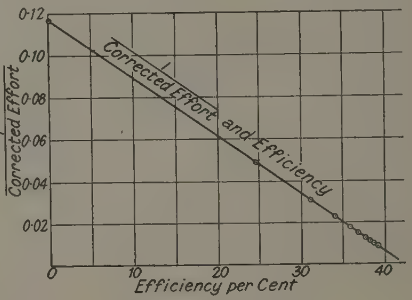


FIG. 119

By *successfully constrained motion* we mean that all movement takes place along a predetermined path; e.g. if we desire a piston to have reciprocating motion only, then we so arrange the connection between the piston and the connecting rod (the gudgeon pin) as to prevent any rotation of the piston about its own longitudinal axis. Now it is found that all constrained motion taking place between any two elements can be classified as, (1) sliding, (2) rolling, or (3) a combination of sliding and rolling. If the connection or *pairing* between any two elements is such that movement results in the *sliding* of one surface over the

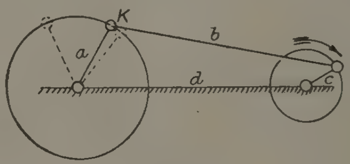


FIG. 120

other, then the pairing is termed *Lower*; if, on the other hand, rolling of the surfaces in contact takes place, then the pairing is termed *Higher*. A screw and nut, a shaft in its bearings, and a piston in its cylinder are typical examples of lower pairing; it will be noticed that contact is made over a *surface* and motion results in the sliding of one surface over the other. Friction and toothed wheels when accurately formed supply us with examples of *higher pairing*; the wheels or teeth, as the case may be, roll over each other and, excluding the slight local elastic flattening of the material in contact, contact is at a point or along a line.

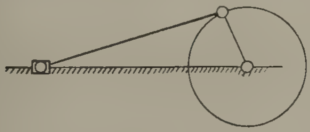


FIG. 121

INVERSIONS OF QUADRIC CYCLE CHAINS

In Fig. 120 the link *d* is seen to be fixed; if, however, we fix one of the other links we obtain an *inverted* mechanism.

SLIDER CRANK CHAIN. The *Quadric Cycle Chain* consists of *four turning pairs*, whilst the *Slider Crank Chain* has *one sliding* and *three turning pairs* and can be obtained from the *Quadric Cycle Chain* by an alteration in the relative lengths of two of the links as follows: Referring to Fig. 120, if link *a* is longer than link *c*, then *c* will make a complete revolution whilst *a* will oscillate through an angle. Further, as the length of link *a* is continually increased, the arc along which the end *K* moves becomes flatter and, will finally become a straight line. The path of *K* can now be set to coincide with the axis of the fixed link *d*, and we now have the usual form of *Slider Crank Chain* (Fig. 121).

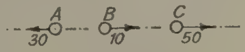


FIG. 122

VELOCITY AND ACCELERATION DIAGRAMS

The *Velocities* and *Accelerations* of various points in a mechanism are conveniently determined by drawing the appropriate velocity and acceleration diagrams. Whilst both velocities and accelerations are *vector* quantities, nevertheless, accelerations are a little more difficult to deal with, so we will first confine our attention to velocities and extend our study to accelerations later.

Let us consider 3 bodies, *A*, *B*, and *C*, Fig. 122, moving along the same straight line, but with the different velocities as shown. We require to show that—

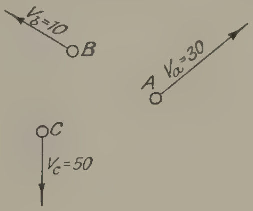


FIG. 123

Velocity of A relative to C = velocity of A relative to B + velocity of B relative to C . (1)

Since *A* and *C* are moving in opposite directions, the velocity of *A* relative to $C = 50 + 30 = 80$. Similarly the velocity of *A* relative to $B = 30 + 10 = 40$. Again, the velocity of *B* relative to $C = 50 - 10 = 40$, since they both move in the same direction. Substituting these values of relative velocities in (1), we have—

$$80 = 40 + 40.$$

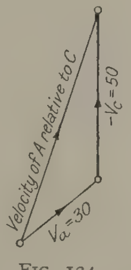


FIG. 124

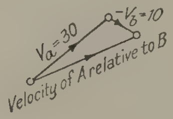


FIG. 125

We can now consider the case of 3 bodies, *A*, *B*, and *C*, Fig. 123, moving with velocities of 30, 10, and 50 respectively, in directions inclined to each other as shown. We have again to show that

Velocity of A relative to C = velocity of A relative to B + velocity of B relative to C . (1)

To determine the *velocity of A relative to C*, we give to C a velocity equal in magnitude but opposite in direction to that which it already possesses; the result of this velocity addition is to cause C to become stationary. Now, in order not to disturb the relative motions, we must serve both A and C alike, that is, we must give to A a velocity equal to that which we have given to C. This is shown in Fig. 124; the vector sum gives the velocity of A relative to C. Again, to determine the *velocity of A relative to B*, we must give to B a velocity equal in magnitude but opposite in direction to that which it already possesses, and so render B stationary. An equal addition of velocity must now be made to A, and so we obtain the *velocity of A relative to B* as shown in Fig. 125. In a similar manner we obtain the *velocity of B relative to C*, Fig. 126. We can now perform the vector sum.

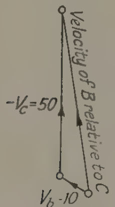


FIG. 126

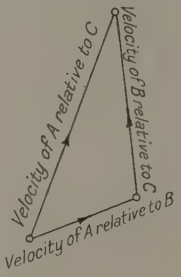

Velocity of A relative to C
Velocity of B relative to C
Velocity of A relative to B

FIG. 127

Velocity of A relative to B + velocity of B relative to C, as in Fig. 127, and measurement shows that the *velocity of A relative to C* as given in Fig. 127 is identical in magnitude and direction with the velocity of A relative to C as given in Fig. 124. As an example on the construction of *velocity diagrams*, let us consider the slider crank chain, Fig. 128, in which the crank rotates with a uniform angular speed w.

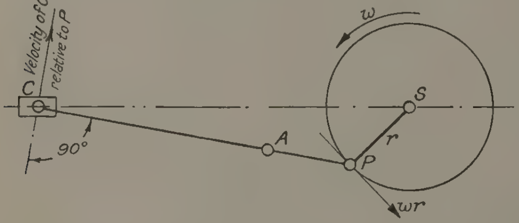


FIG. 128

Setting down our fundamental equation, we write: *Velocity of slider C relative to the fixed line CS = velocity of slider C relative to crank pin P + velocity of crank pin P relative to the base line CS.*

Now, the velocity of P is wr tangentially to the crank circle at P; set down sp, Fig. 129, $= wr$ to scale. Also, the velocity of C can only be directed along the line CS, therefore draw sc parallel to SC.

In considering the velocity of C relative to P, we must realize that C and P are the extremes of a *rigid* rod and, therefore, neither end can have a velocity directed towards the other. In reality, CP is a rod which we may consider to be pinned and capable of swinging about P, so that the velocity of C relative to P is directed along a line at right angles to CP, as shown in Fig. 128. Returning to Fig. 129, draw cp perpendicular to CP, then cp represents the velocity of the slider C relative to P, and sc represents the velocity of the slider C relative to S, that is, along the guides. If it is required to find the velocity of any point A on the connecting rod CP, we locate the point a, so that $\frac{ca}{cp} = \frac{CA}{CP}$; then sa represents the velocity of this point in magnitude and direction relative to the fixed link or line CS.

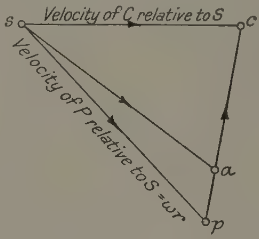


FIG. 129

ACCELERATION DIAGRAMS

The fundamental equation for accelerations is of similar form to that for velocities and is written—

Total acceleration of slider C relative to fixed line CS = total acceleration of slider C relative to P + total acceleration of P relative to the fixed line CS.

Now, the *total* acceleration is the vector sum of the tangential and centripetal accelerations, and we must, as far as possible, determine the tangential and centripetal accelerations of, (1) slider C relative to CS, (2) slider C relative to P, and (3) P relative to S, before we can draw the acceleration diagram. (1) Since slider C moves wholly in a straight line it can have no centripetal (at right angles to CS) acceleration; its whole acceleration must be directed along CS and, at present, is unknown in magnitude. (2) The centripetal acceleration of C relative to P is directed from C towards P, and its magnitude is $\frac{(cp)^2}{CP}$ (cp, Fig. 129, is the linear velocity of C when we consider it as moving about P as a centre). The tangential acceleration of C is unknown at present, except that it is directed at right angles to CP. (3) The tangential acceleration of P is zero since the speed of rotation is constant. The centripetal acceleration of P is directed towards S, and its magnitude is $\frac{(sp)^2}{SP}$. We now have sufficient data to enable us to draw the acceleration diagram, Fig. 130, for this configuration of the chain. Set down, $s^1p^1 =$ centripetal acceleration of P towards $S = \frac{(sp)^2}{SP}$. From p^1 draw p^1c_p parallel to CP and $=$ centripetal acceleration of C relative to $P = \frac{(cp)^2}{CP}$. The tangential acceleration of C relative to P is unknown in magnitude but its direction is at right angles to CP, or p^1c_p; therefore draw c_pc^1 at right angles to p^1c_p. The tangential acceleration of C relative to S is unknown in magnitude, but its direction is along CS, therefore draw s^1c^1 parallel to CS. Then, c^1s^1 represents the tangential acceleration of C relative to S, and is seen to be the vector sum of s^1p^1 (which is the total acceleration of P relative to S) $+ p^1c^1$ (which is the total acceleration of the slider C relative to P).

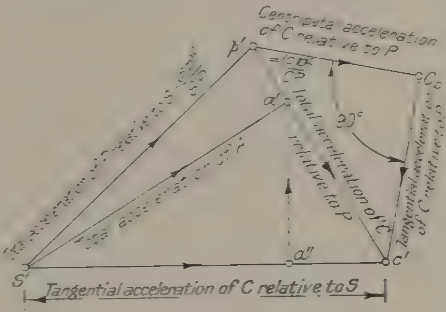

Centripetal acceleration of C relative to P
p'
$= \frac{(cp)^2}{CP}$
C_p
a'
total acceleration of C relative to P
90°
tangential acceleration of C relative to P
a''
c'
S'
Tangential acceleration of C relative to S

FIG. 130

Having located a^1 to satisfy the relation $\frac{c^1a^1}{c^1p^1} = \frac{CA}{CP}$, then s^1a^1 is the total acceleration of A relative to the fixed link or base CS. If $a'a''$ is drawn perpendicular to s^1c^1, then $s'a''$ and $a'a''$ represent the accelerations of A in directions parallel and normal respectively to CS.

PIONEERS OF ENGINEERING

By J. F. Corrigan, M.Sc., A.I.C.

LESSON XXI

MEASUREMENT STANDARDS ESTABLISHED

SIR JOSEPH WHITWORTH, Bart.

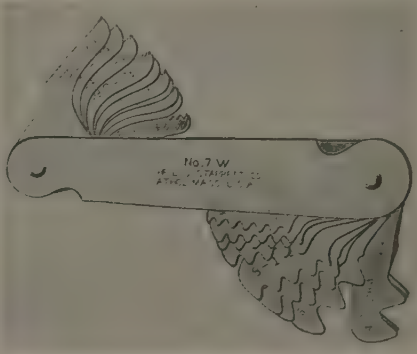
(1803–1887)

The great handicap of all the earlier steam-engine inventors and mechanical engineers was that they have no definite constructional standards of measurement to refer to. Watt, in his writings, very frequently is found lamenting the lack of such definite measuring standards. Watt and his fellow engineers had to deal with engines in which the nuts, bolts, rods, and the other items of assemblage were all of different sizes, and, as the reader will well be able to imagine, the repair of engines and machinery in those days was a task which called for the greatest amount of patience, owing to the fact that each extra nut and bolt needed had to be made specially to fit the required part.

Sir Joseph Whitworth

In the beginning of the nineteenth century, however, there was born an individual who was destined to remedy the above state of affairs, and, by his insistence upon definite standards of reference in mechanical engineering, to place the whole of the engineering industry on a greater and a more productive basis than it had occupied before.

Joseph Whitworth, for such was the name of this apostle of engineering accuracy, was born at Stockport, near Manchester, on 21st December, 1803. Like most youths of his time, he left school at the age of 14, and was apprenticed to a local firm of cotton spinners. Such a task, however, did not satisfy young Whitworth. He gave up the job, and went to London. After a period of various engagements in that city, he returned to Manchester, and, with the aid of capital which he had influenced, he set up his own engineering works, the main line of the business being tool making.



Whitworth Screw-pitch Gauge

It was about this time that Whitworth realized the vital importance of setting up accurate standards of measurement, and from this period onwards one of the chief ends of his business life was to create the necessary measuring standards, and, still more, to convince other people of their practical value.

Whitworth read many papers before the British Association and the Institute of Civil Engineering. It was in a paper read before the latter body in 1841 that Whitworth urged the adoption of a comprehensive and uniform system of screw-threads. He began to manufacture standardized screw-threads in his own works. Gradually, his system of standardized screw-thread gauges came into universal use, and now, of course, at our present time, the Whitworth thread gauge is a system of reference in engineering measurements which is world famed.

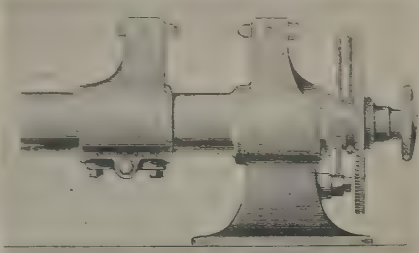
In addition to creating the above standards, Whitworth devised other important methods for making accurate measurements. For instance, he showed that if any two of three surfaces fit together perfectly, all the three surfaces must be true planes. He invented also a machine for making very minute measurements of length, and, by means of some of the present-day Whitworth measuring machines, it is possible to effect practical measurements of an order of $\frac{1}{1000000}$ in.

WHITWORTH SURFACE PLATES

Whitworth, at the request of the Government, investigated the subject of the rifling of gun barrels. He invented new and improved machines for making rifles and heavy guns in large quantities. Another process originally due to him is a method of compressing steel in its molten state, whereby a metal of very great tensile strength is obtained.

In 1868, Whitworth founded the scholarships which are now named after him. He set aside an annual sum of £3,000 to be awarded in scholarships for "intelligence and proficiency in the theory and practice of mechanics and its cognate sciences."

The following year Whitworth was created a Baronet. He died in 1887, and in his will he left the sum of £100,000 for the permanent endowment of Whitworth scholarships. It is estimated that, during his lifetime, Sir Joseph Whitworth expended no less than half a million pounds on gifts to local educational bodies, and many of these organizations benefited in a permanent manner after his death.



A WHITWORTH MEASURING MACHINE

(This instrument reads to $\frac{1}{20000}$ inch)

Sir Joseph Whitworth, Bart., without a doubt, left his footprints on the sands of time. His educational bequests have conferred a lasting benefit upon his fellow men, but more than ever will his name be perpetuated by his creation of the engineering standards of measurement which are now employed to such a wide extent in the engineering industry.

STEAM CONDENSING PLANT

By John Evans, M.Eng.

LESSON IX

COOLING TOWERS AND SPRAY PONDS

An unfailing and copious supply of cold and reasonably clean water is necessary for the economic operation of all types of modern condensing plant. Although a natural supply is advantageous, it is not always suitable, as it may contain chemicals and other impurities which would injure the metals of which the condensing plant is constructed. In such a case, and where a clean natural supply is strictly limited or not available, cost prohibits the use of water from the mains unless it can be cooled and used over and over again.

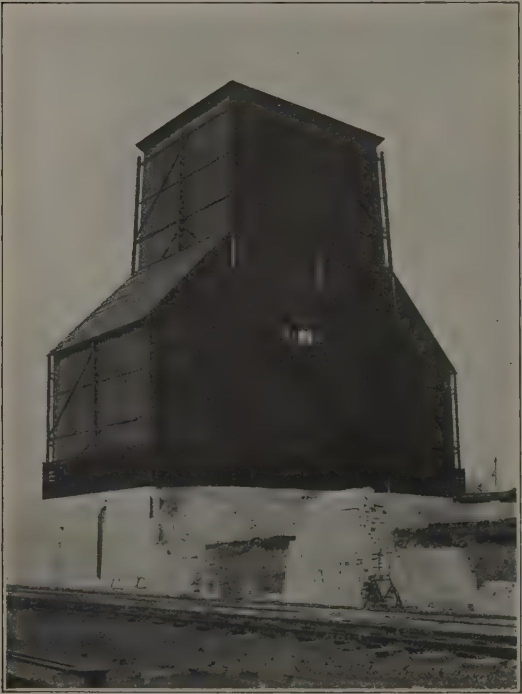


Fig. 45. Natural Draught Cooling Tower

The two most common cooling devices are—

(*a*) Cooling towers.

(*b*) Cooling or spray ponds.

In both, the necessary cooling is produced by dividing up the water into small drops or a spray, and allowing it to fall through the air into a reservoir or pond. Evaporation of a small percentage of the water, (1 to 2 per cent, depending on conditions), results in a lowering of the temperature of the main body of the water. Each pound of water evaporated absorbs about 1,000 heat units from the main body of units, which is equivalent to cooling 50 lb. of water 20° F.

Cooling Towers. Cooling towers are generally of two types—

(*a*) Natural draught.

(*b*) Forced draught.

They consist of a wooden, steel, or concrete housing, open at each end, and filled with wooden, steel, or concrete laths, so arranged that the water to be cooled is discharged at the top, and distributed so that it falls in thin sheets or as spray, and trickles down the interior obstructions to a tank or reservoir at the bottom. Air flows up the tower, through the dripping water, and rapidly cools it by evaporating a small portion.

The sides of the natural draught tower (Fig. 45) are closed, except for louvred openings at the bottom through which the air enters the tower, and a closed flue or chimney is erected over the portion containing the cooling stack to produce the necessary draught.

The sides of forced draught towers are entirely closed, and in place of the louvred openings at the bottom are openings for fans which provide the necessary air.

A typical cooling stack is shown in Fig. 46. It is carried on piers, and consists of a large number of laths of square section fixed so that the diamond points are set uppermost. Water is delivered to the water distributor *A*, and supplied to the distributing troughs *C* by intermediate troughs *B*. The distributing troughs *C* are totally enclosed, and spraying nozzles fixed at definite intervals along the tops, spray the water over the stack.

The natural draught tower, although larger than the forced draught type, requires no power for its operation. In the latter type 1 to 2 per cent of the power output may be required for driving the fans. On the other hand, the forced draught type is continuous in operation, and independent of air temperature and winds.

The steel shell tower (Worthington, Simpson) has much to recommend it. It has a longer life than the wooden tower, occupies less space, is not so easily susceptible to the action of the weather, and the risk of damage by fire is negligible.

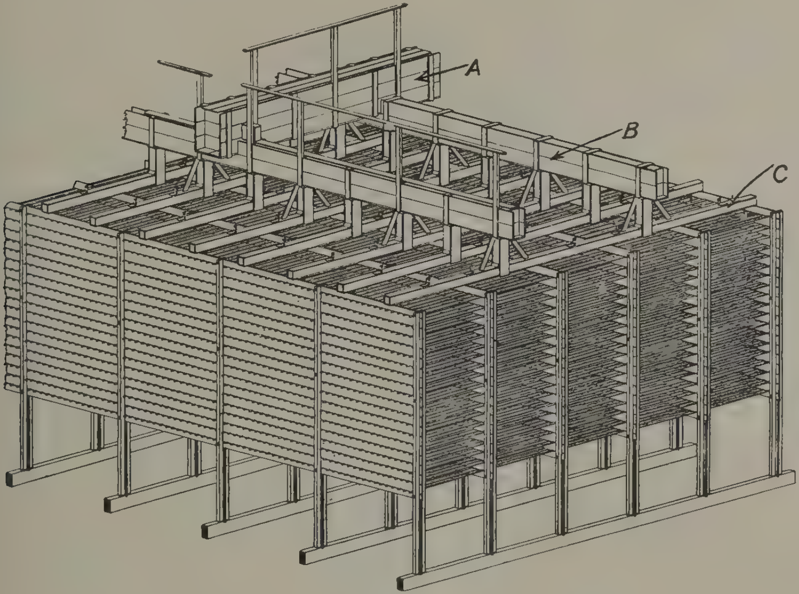


Fig. 46. Diagram Illustrating the Irrigation

The apparatus for distributing and breaking up the water (Fig. 47) is located in the interior, and consists of a star-shaped rotary water distributor mounted on the top of a stand pipe through which the water to be cooled is admitted. Rotation of the distributor is maintained by the reactionary pressure, due to the water issuing in jets through a series of orifices on the main distributing arms.

The breaking up of the jets into fine drops and exposing them to the current of cool air is effected by many thousands of thin galvanized steel sheets, rolled into tubes and arranged to interlock.

Sections of natural draught and forced draught steel shell towers are shown in Figs. 48 and 49 respectively.

Example. A natural draught cooling tower is required to cool 100,000 gal. of water per hour from 90° F. to 65° F. The temperature of the atmosphere is 55° F., and the relative humidity 85 per cent. Barometer, 30 in. Hg.

Air Entering Tower. Pressure of dry saturated vapour at 55° F. (steam tables) = 0·4357 in. Hg. Since the relative humidity of the atmosphere = 85 per cent, i.e. the air is 85 per cent saturated with water vapour, the pressure of vapour in the air = 0·85 × 0·4357 = 0·3704 in. (Dalton's law).

$\therefore$ Pressure of the dry air = 30 − 0·3704
= 29·6296 in. Hg.

If T = absolute temperature of air ° F.
P = absolute pressure of air lb.-sq. ft.

Volume per pound of dry air
= volume of vapour per pound of air

$$= \frac{RT}{P} = \frac{53\cdot2 \times (460 + 55)}{144 \times 0\cdot49 \times 29\cdot6296} = 13\cdot10 \text{ cub. ft.}$$

Now the volume of the molecular weight of any gas or vapour at standard temperature and pressure is constant, and if the molecular weight is expressed in pounds, the molecular volume is 357 cub. ft.

The molecular weight of water vapour is 18, so that 18 lb. at a pressure of 30 in. of mercury, and a temperature of 32° F. occupy 357 cub. ft.

∴ Weight of 1 cub. ft. of vapour at 55° F. and a pressure of 0·3704 in. Hg.

$$= \frac{18}{357} \times \frac{0{\cdot}3704}{30} \times \frac{460 + 32}{460 + 55}$$

$$= 0{\cdot}000595 \text{ lb.}$$

∴ Weight of vapour per pound of air

$$= 0{\cdot}000595 \times 13{\cdot}10 = 0{\cdot}0078 \text{ lb.}$$

If Kp = specific heat of air = 0·238
t° F. = temperature of air and its associated vapour entering tower
L = latent heat of vapour at temperature t° F. = $1114 - 0{\cdot}7t$.

		B.Th.U.'s
Then heat carried into tower by the air $= Kp(t - 32) = 0{\cdot}238(55 - 32)$	=	5·47.
and heat carried into tower by the vapour $= 0{\cdot}0078\{t - 32) + L\}$ $= 0{\cdot}0078\{55 - 32 + 1114 - 0{\cdot}7 \times 55\}$	=	8·57
Total heat carried into tower per pound of air and its associated vapour	=	14·04

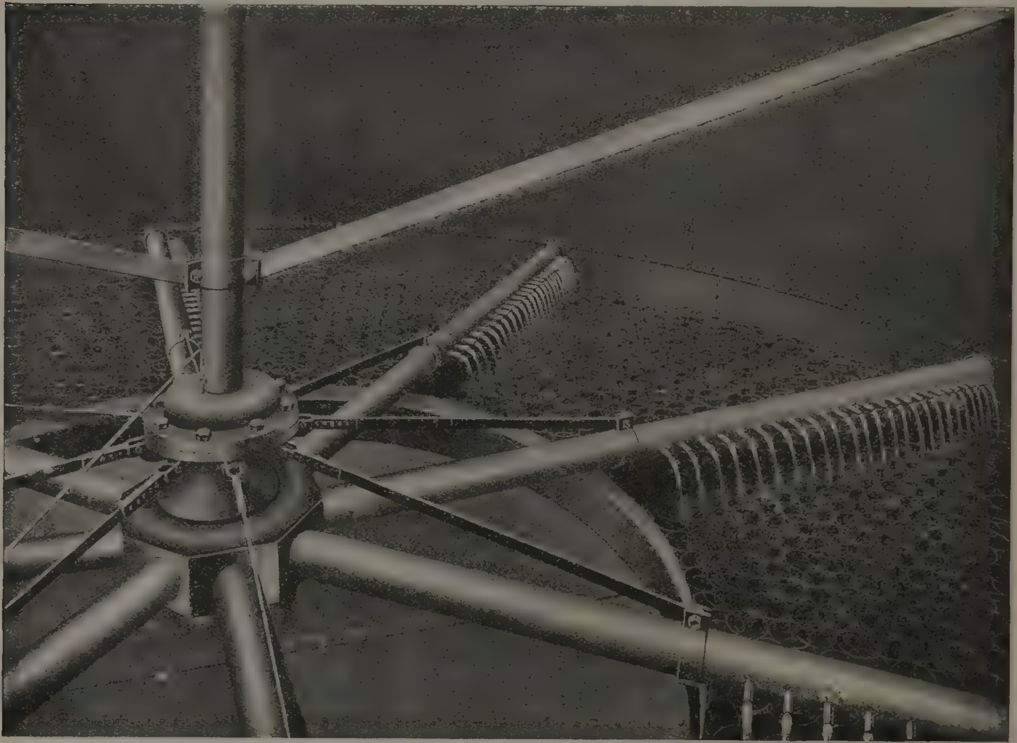


FIG. 47. INTERIOR. SHOWING ACTION OF ROTARY DISTRIBUTOR AND FILLING

Air Leaving Tower. Assume that the air leaves the tower at a temperature 10° F. lower than that at which the water enters, = 90 − 10 = 80° F., and its relative humidity is increased to 90 per cent.

Pressure of dry saturated vapour at 80° F. (steam tables) = 1·029 in. Hg.

Pressure of vapour in the air = 0·9 × 1·029 = 0·9261 in.

Pressure of dry air = 30 − 0·9261 = 29·0739 in.

Volume per pound of dry air

= volume of vapour per pound of air

$$= \frac{53{\cdot}2 \times (460 + 80)}{144 \times 0{\cdot}49 \times 29{\cdot}0739} = 14{\cdot}00 \text{ cub. ft.}$$

Weight of 1 cub. ft. of vapour at 80° F. and a pressure of 0·9261 in. Hg.

$$= \frac{18}{357} \times \frac{0{\cdot}9261}{30} \times \left(\frac{460 + 32}{460 + 80}\right)$$

$$= 0{\cdot}001412 \text{ lb.}$$

∴ weight of vapour per pound of air

$$= 14{\cdot}00 \times 0{\cdot}001412 = 0{\cdot}0197 \text{ lb.}$$

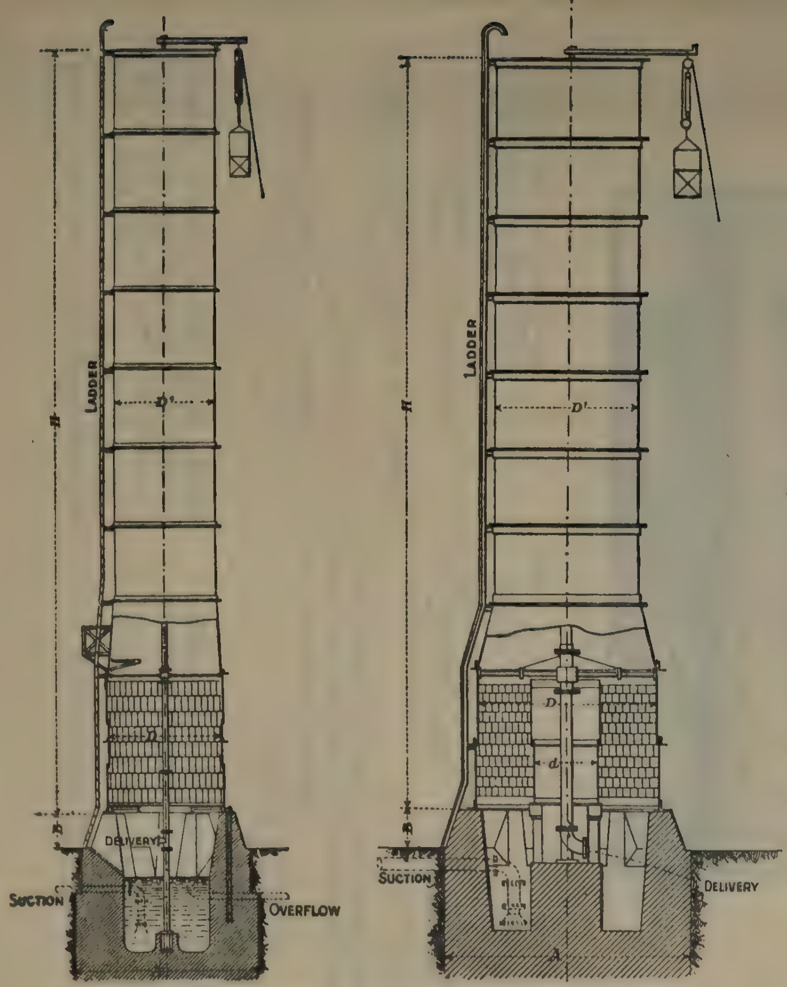
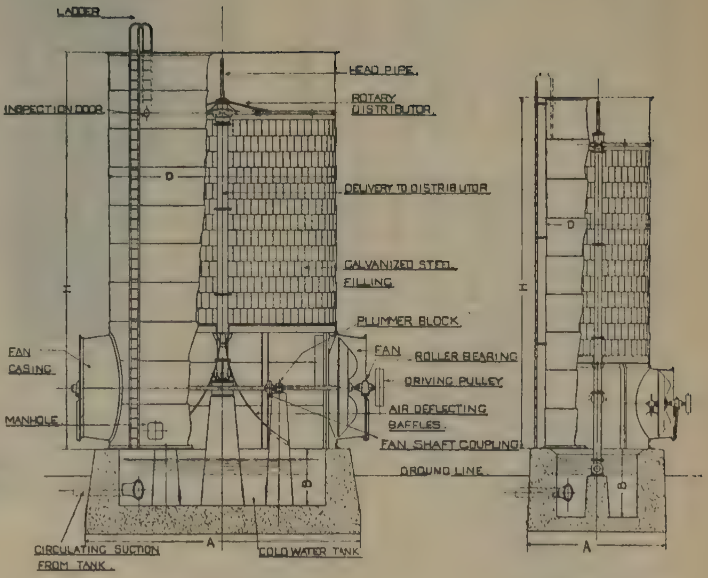


FIG. 48. SECTIONAL VIEWS OF NATURAL DRAUGHT COOLING TOWERS

Section of Natural Draught Cooling Tower less than 18 ft. diameter

Section of Natural Draught Cooling Tower more than 18 ft. diameter



MULTIPLE FAN TYPE.

SINGLE FAN TYPE.

FIG. 49. WORTHINGTON-SIMPSON FORCED DRAUGHT COOLING TOWER

		B.Th.U.'s
Heat carried away from tower by the air $= 0{\cdot}238(80 - 32)$	=	11·42
Heat carried away from tower by the vapour $= 0{\cdot}01977\{80 - 32 + 1114 - {\cdot}7 \times 80\}$	=	21·86
∴ total heat carried away from tower per pound of air and its associated vapour	=	33·28

Weight of Air Required. Total heat to be absorbed by the air in flowing through the tower

= weight of water cooled per hour × temperature drop

$= 100{,}000 \times 10 \times (90 - 65) = 25{,}000{,}000$ B.Th.U.'s per hour.

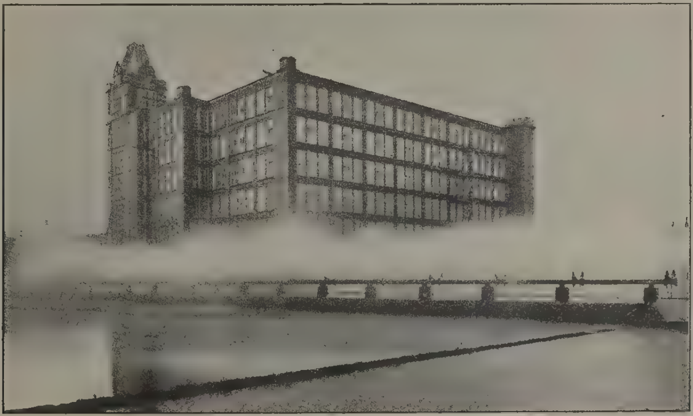


FIG. 50. SPRAY COOLING POND

Heat absorbed per pound of air

$= 33{\cdot}28 - 14{\cdot}04 = 19{\cdot}24$ B.Th.U.'s

∴ total weight of air required per hour

$$= \frac{25{,}000{,}000}{19{\cdot}24} = 1{,}300{,}000 \text{ lb.}$$

Increase in weight of vapour per pound of air in flowing through tower

$= 0{\cdot}01977 - 0{\cdot}00780$

$= 0{\cdot}01197$ lb.

∴ weight of cooling water vaporized

$= 1{,}300{,}000 \times 0{\cdot}01197$

$= 15{,}561$ lb. per hour

$= 1{\cdot}56$ per cent.

Draught. Assume that the height of the tower is 100 ft., and the length of the chimney portion is two-thirds of the total height $= \frac{2}{3} \times 100 = 65$ ft. say.

The temperature of the air inside the tower is greater than that of the atmospheric air outside, and consequently the pressure exerted at the base by the air inside, is correspondingly less than the pressure exerted by an equal column of atmospheric air. This pressure difference is termed draught, and being small, is usually expressed in inches of water.

Weight of 1 cub. ft. of air outside the tower $= \frac{1}{13{\cdot}10}$	= 0·07633 lb.
Weight of 1 cub. ft. of vapour outside the tower	= 0·000595 lb.
∴ weight of 1 cub. ft. of air and associated vapour	= 0·076925 lb.

Weight of 1 cub. ft. of air leaving the tower $= \frac{1}{14{\cdot}00}$	= 0·07143 lb.
Weight of 1 cub. ft. of vapour leaving the tower	= 0·001412 lb.
∴ weight of 1 cub. ft. of air and associated vapour	= 0·072842 lb.

The weight of 1 cub. ft. of air and vapour inside the tower may be taken as the mean of that entering and leaving

$$= \frac{0{\cdot}076925 + 0{\cdot}072842}{2}$$

$= 0{\cdot}074883$ lb.

∴ difference in weight per cub. ft.

∴ $= 0{\cdot}076925 - 0{\cdot}074883 = 0{\cdot}002042$ lb.

$$\text{Draught} = 65 \times 0{\cdot}002042 \times \frac{12}{62{\cdot}4}$$

$= 0{\cdot}0255$ in. of water.

Let p = draught in pound per square foot.
V = volume of 1 lb. of air outside tower
v = velocity of air in chimney, ft.-sec.
C = coefficient of velocity of air = 0·65 for natural draught towers.

Then

$$\begin{aligned} v &= C\sqrt{2gpV} \\ &= 0{\cdot}65\sqrt{2 \times 32{\cdot}2 \times 0{\cdot}002042 \times 65 \times 13{\cdot}10} \\ &= 6{\cdot}88 \text{ ft. per sec.} \end{aligned}$$

$\therefore$ cross-sectional area of chimney

$$= \frac{\text{lb. of air per hour} \times \text{volume per lb. air}}{3600 \times \text{velocity of air in feet per sec.}}$$

$$= \frac{1{,}300{,}000 \times 13{\cdot}1}{3600 \times 6{\cdot}88} = 687 \text{ sq. ft.}$$

Dimensions of chimney portion

65 ft. × 26 ft. × 26 ft., say.

Assuming that the filling material occupies 50 per cent of base of the tower, then

dimensions of base = 35 ft. × 52 ft. × 26 ft.

Spray Ponds. The oldest form of cooling device is the open pond into which warm water discharged from the condenser is cooled, partly by the atmosphere in contact with the surface of the pond (radiation and conduction), but chiefly by evaporation. Air under normal atmospheric conditions is seldom saturated, and as it passes over the pond, it becomes saturated consequent on its rise in temperature, and corresponding greater vapour absorbing capacity. The vapour absorbed by the air is obtained from the water in the pond, the water temperature being thereby lowered. With a slight breeze, air circulation is increased, and a higher degree of cooling obtained. It is evident that the greater the area of the pond, the greater will be the degree of cooling, and while depth does not affect the rate of cooling, it is advantageous, because the deeper the pond, the cooler will be the supply of water drawn from the bottom and returned to the condenser.

Where a large surface is not available, the water is distributed over the pond through pipes and through nozzles, falling to the pond surface as a fine spray (Fig. 50). This facilitates evaporation as the water is atomized, but in addition to the loss due to vaporization, a considerable percentage is carried away in suspension in the air, especially when high winds prevail. The pumping power required is low, the head of water on the nozzles being usually about 20 ft.

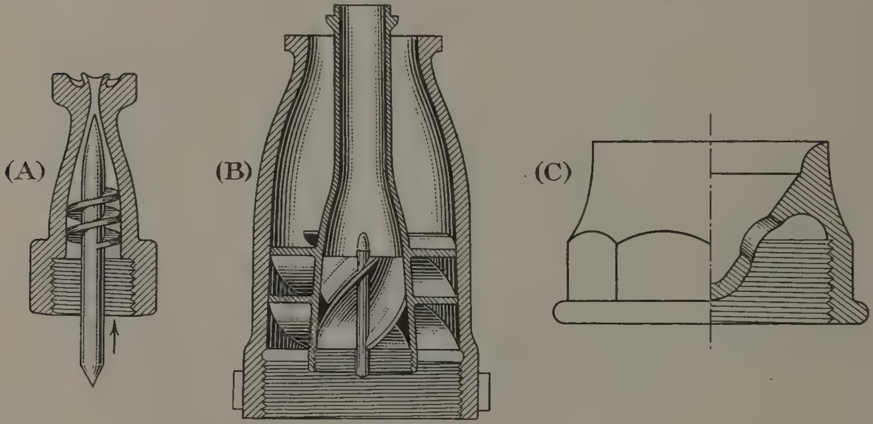


FIG. 51

(*A*) Spray nozzle. (*B*) Low pressure multi-spray nozzle. (*C*) Impingement type of spray nozzle.

Spray nozzles of the centrifugal and impingement types are shown in Fig. 51. They are made of gun-metal and screwed on the distributing pipes at suitable intervals.

Type *A* has special vanes fixed on the nozzle spindle. These vanes cause rotation of the water which is atomized by centrifugal action. The multi-spray type *B* consists of one nozzle within another, the double spiral arrangement producing two distinct sprays. With the third type *C*, atomization of the water is brought about by the impingement together of pairs of plain jets.

(CONCLUSION)

STRUCTURAL STEELWORK

By W. ARNOLD GREEN, M.A., B.Sc., A.M.Inst.C.E., M.I.Struct.E.

LESSON IV

STRUCTURAL STEEL—(*contd.*)

Rivet Pitch in Girder Flanges. The required pitch of the rivets at any point depends on the shear.

If in the two plated joists discussed in the last lesson, the maximum end shear is 50 tons, the shear stress on the 13 in. width between the joists and the plate equals $S \times A \times a \div t \times I$, where the symbols have the meaning assigned to them on page 824, Lesson I.

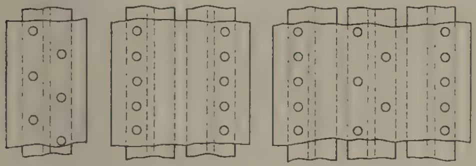


FIG. 21

The shear per inch run is thus $S \times A \times a \div I = 50 \times 16 \times \frac{1}{2} \times 10\frac{1}{4} \div 3654 = 1{\cdot}12$ tons per 1 in. At 5 tons per sq. in. the value of two $\frac{7}{8}$ in. rivets was shown on page 916, Lesson II, to be 6·01 tons—the required pitch is thus $6{\cdot}01 \div 1{\cdot}12 = 5{\cdot}36$ in., say, 5 in. pitch.

In fastening the plates to one, two, or three joists the usual arrangement of rivets is as shown in Fig. 21.

In the net section one rivet hole is omitted from each flange of each joist. In the compression flange, if the rivets completely fill the holes as they should, it is reasonable to deduct nothing for holes, but for simplicity it is usual to calculate on the assumption that the girder is symmetrical about a central horizontal axis, so that the stress in the compression flange is less than the stress assumed.

Plate Girders. Calculations for plate girders of built-up sections are sometimes made by the section modulus method explained above, but where different working stresses are specified for tension and compression the top and bottom flanges may be of different section.

The total load on each flange at any point is taken as the bending moment divided by the distance between the centroid of each flange, in the area of which it is usual to include one-eighth of the area of the web, provided that the cover plates at the web joints are of sufficient area and have enough rivets to transmit across the joint that portion of the flange load which the web is assumed to carry.

It is usually sufficiently accurate to assume a constant distance between the centroids of the flanges, so that the bending moment diagram to another scale will be the required diagram of flange area of which the flange angles or sections connected directly to the web should form a large proportion of the whole area.

The calculation for the rivets required for connecting the flange plates to the angles can be made as for compound girders, but the pitch is usually made the same as that required for connecting the web to the angles. B.E.S.A., No. 153, specifies that enough rivets must be provided at the end of the plates to transmit the whole of the load they are assumed to carry, half of which must be placed beyond the theoretical stopping off points.

If, to connect the web to the angles, a single line of rivets is used each of value R tons at pitch p inches, and the total horizontal shear coexisting with a vertical shear V is transmitted by these rivets from the web to the flange then for equilibrium to be maintained, $R \times d = V \times p$, where d is the vertical distance in inches between the rivets in each flange. (See Fig. 22.)

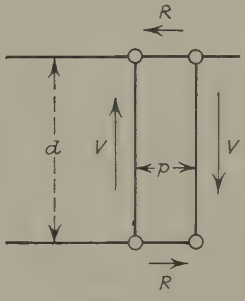


FIG. 22

If one-eighth of the web area acts with the flanges, the stress in that portion is not transmitted from the web by rivets, so that the value of V in the above formula may be reduced in the ratio of the flange area to the sum of the flange area and one-eighth of the web area.

In addition to transmitting the horizontal shear, these rivets have to transmit to the girder the load bearing on the top flange.

If, in the last formula, p equals 12 in., R will equal the tons per foot run to be transmitted horizontally by the rivets. This must be combined with the tons per foot run (say W) transmitted vertically, so that the total load per foot run is $\sqrt{R^2 + W^2}$. Twelve times the value of one rivet divided by this value gives the required pitch of the rivets in inches.

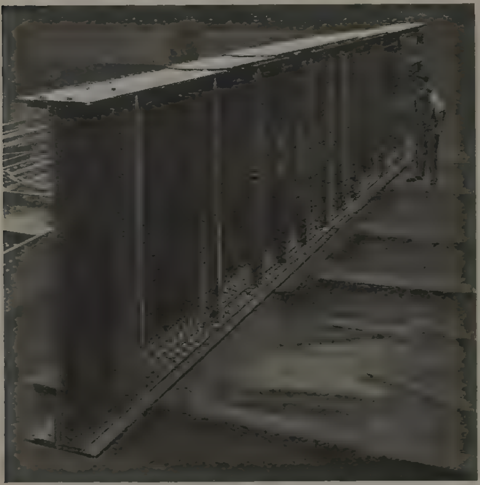


FIG. 23. SINGLE WEB PLATE GIRDER

The half-tone blocks, Figs. 23 and 24, are from photographs kindly lent by Messrs. Dorman, Long & Co., Ltd. They show two heavy plate girders in their shops, one with a single web and the other with three webs. The stopping of the flange plates and the stiffeners fitting between the flange angles are clearly shown.

Impact. In ordinary building construction the loads coming on the floors are assumed equivalent to dead loads, which are added to the permanent weight of the structure itself, including floor finishes, walls, etc. Except in special cases, these dead loads may be taken as including an allowance for the impact effect due, e.g. to movement of people, the sudden dropping of a bale, or the vibration of machinery.

For girders carrying rolling loads, such as travelling cranes, traction engines, trams or railway trains, the effect produced on various parts of the structure is obviously greater than stationary loads would produce. This increased effect is sometimes allowed for by specifying different working stresses in various parts of the structure, varying with the sort of loading and the span of the girder.

When a train crosses a railway bridge, the unbalanced driving wheels of the engine, the unevenness of the track or wheel tyres, the swaying from side to side, the quickness of application of the load, and the deflection of the girders directly under the wheels, all contribute to the extra stresses in the structure.

Many formulae have been devised in an attempt to estimate the probable increased effect of the rolling load due to impact; e.g. the late Mr. C. C. Schneider's impact coefficient is $\frac{300}{300 + L}$ where L is the length (in feet) of loaded track producing the maximum stress in the member considered. The equivalent dead load in the member contributed by the rolling load is the load produced by the rolling load as if stationary, in the position producing the maximum effect, multiplied by $1 +$ the impact coefficient.

Mr. J. A. L. Waddell's coefficient is $165 \div (150 + nL)$ for railway bridges and $100 \div (200 + nL)$ for highway bridges where n is the number of tracks carried by the railway bridge, or the total width of the bridge (in ft.) divided by 20 for highway bridges.

The tentative value of the coefficient given in B.E.S.A., No. 153, is $120 \div [90 + (n + 1) L/2]$, where n is the number of tracks the member is designed to support or assist in supporting.

The value of L for the rail bearers is to be taken as their effective span and for a cross girder twice this amount, i.e. the distance between the cross girder centres on each side of the member in question. The value of the coefficient is to be limited to 1·15, and may be reduced for varying spans and speeds.

For road bridges, the impact coefficient is two-thirds the above, where n is the number of lines of traffic carried by the member considered. The American Railway Engineering Association from a series of tests has devised the impact coefficient $1 \div (1 + L^2/20{,}000)$ where L is the span in feet of the bridge or rail bearers, or the length of two panels of a bridge for the cross girders.

Fatigue. If the stress in a bar of steel alternates between zero and a maximum, it will fail under repeated applications of the load if the maximum stress exceeds about half the load that would break it in one application. If the alternation is from a tension to a compression of the same value, the bar will fail if the stress exceeds about one-third of the ultimate stress.

If the calculated maximum and minimum stresses are those due to bending of the specimen, the endurance limit (that is, the largest stress that will not cause failure by any number of repeated applications) is higher than for direct stresses. With the stresses normally used in structural steelwork there should be no danger of failure by *fatigue*, as this phenomenon is termed.

For members subject to reversal of stresses it is usual, however, to design their connections for the larger load plus a proportion of the smaller. B.E.S.A., No. 153, requires the member itself to have a gross area for the larger stress plus half the gross area required for the smaller.

Wind. From experiments at the National Physical Laboratory, it has been found that the pressure in lb. per sq. ft. on a surface normal to the direction of the wind equals approximately the square of the velocity of the wind in miles per hour multiplied by 0·0032. As 100 miles per hour is regarded as the velocity of the wind in a violent hurricane, a pressure of more than 30 lb. per sq. ft. except in very exposed positions need rarely be allowed for in this country. In sheltered positions, a considerably less pressure may be expected.

On surfaces, such as roof slopes oblique to the direction of the wind, usually taken as horizontal, only the normal pressure is considered. This is greater than the normal component of the horizontal pressure on a vertical surface and may be taken from Duchemin's formula, as equal to the horizontal pressure on a vertical surface multiplied by $\frac{2 \sin \theta}{1 + \sin^2 \theta}$ where $\sin \theta$ is the ratio of the rise to the length of slope.

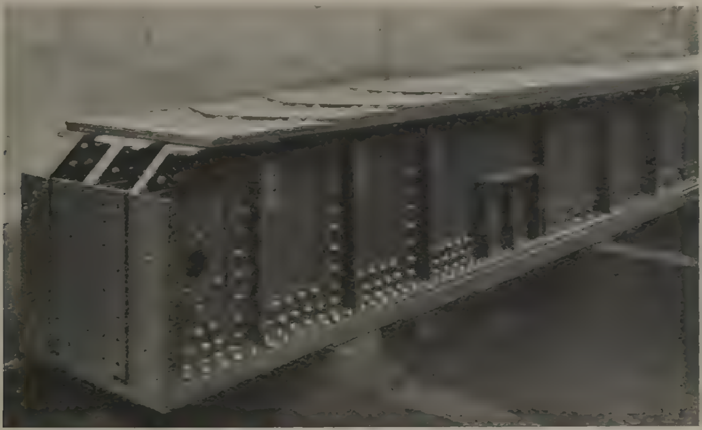


FIG. 24. PLATE GIRDER WITH THREE WEBS

Wind also exerts a suction on the leeward side, taken in the Belgian standard specification as an uplift of ·7 of the standard pressure (much smaller than that given by Duchemin's formula) on roofs resting on walls.

The treatment for wind on steel frame buildings and bridges is discussed in the next lesson.

MACHINE TOOLS

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XVIII

MILLING MACHINES—(*contd.*)

A much modified design of machine built on the Lincoln model is the automatic milling machine, Fig. 126, by the Brown & Sharpe Mfg. Co., of Providence, Rhode Island, U.S.A. It is a modern design, differing from the older forms in having a constant speed drive, which permits of effecting changes for spindle and table, independently of each other, with a wide range of table feeds for any spindle speed, and vice versa. The single pulley drives through a friction clutch, which, when stopped, operates a brake to stop the machine instantly. A motor can be substituted for the belt pulley. The leading features of this machine are as follows: The friction clutch pulley runs on ball bearings at 350 r.p.m. Twelve changes of spindle speeds are obtained in geometrical progression with gears, from 22 to 180 r.p.m. It is driven (Fig. 127) through a silent chain, sprockets and spur gearing. The spindle has a vertical adjustment of 15 in. The table has a working surface of 52 in. by 12½ in. surrounded with an oil pan. It has a transverse adjustment of 5¼ in. and a longitudinal feed of 34 in., with 18 changes through gears, and independent of spindle speeds. The changes range from ·38 in. to 24·54 in. per minute, the range for small mills is ·002 in. to ·136 in. per revolution of spindle, for large mills from ·017 in. to 1·115 in. per revolution of spindle. A fast travel between cuts of 210 in. per minute is provided. The movements are controlled by hand with levers in front of the saddle, or automatically with dogs on the front of the table.

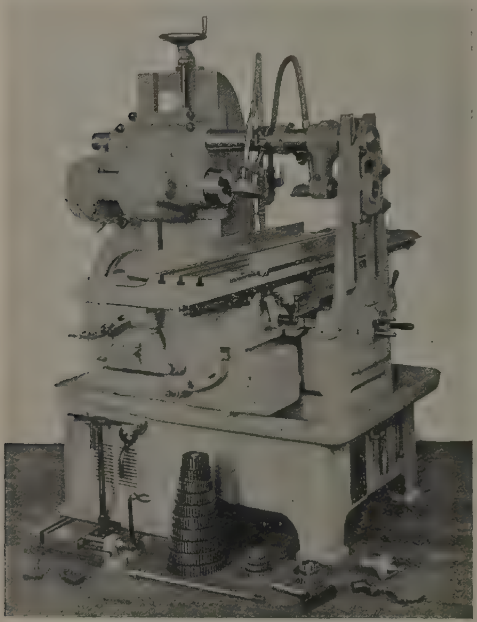


Fig. 126. Modified Lincoln Type of Milling Machine

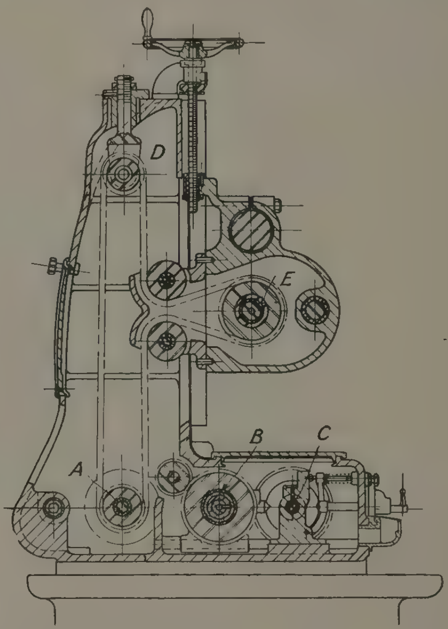


Fig. 127. Spindle Drive of Previous Machine

The method of driving is outlined in Fig. 127, the outlines of which may be compared with the photograph. The drive commences with the shaft *A*, actuated by the driving pulley, and friction clutch at 350 r.p.m. This drives shaft *B* through spur gears, on which are mounted automatically-operated friction clutches that provide for the starting, stopping, and reversal of the spindle. Power is delivered to the spindle with a silent chain, the course of which is outlined in Fig. 127, from a sprocket mounted on a sleeve running loosely on the shaft *A*, and driven from shaft *B* by the automatically-operated friction clutches. Shaft *C* carries the cams that control the automatic operation of the stopping, starting, and reversing friction clutches, the shaft *C* being driven from *B* with spur gears, and their operation is controlled by the hand levers, and table dogs mentioned. The chain runs over the adjustable sprocket *D* at the top of the column over a pair of idlers to the spindle driving sprocket *E*, shown in Fig. 128. The sprocket *D* is adjusted from the outside of the machine to maintain the proper tension on the chain.

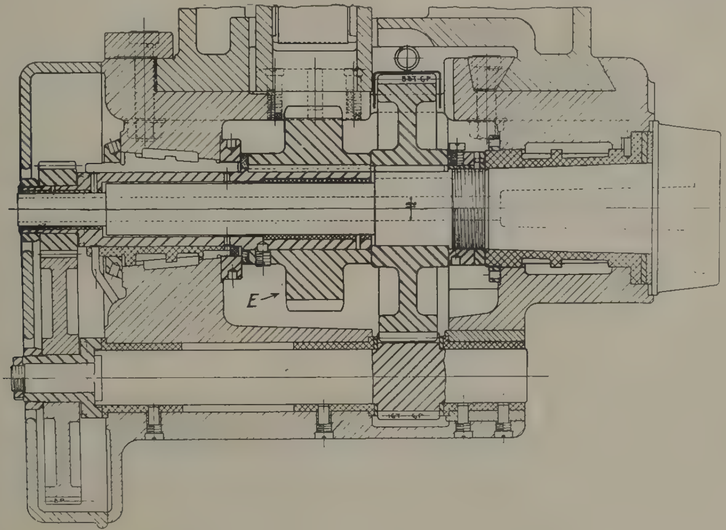


FIG. 128. DETAILS OF SPINDLE ARRANGEMENTS

The section through the spindle head, Fig. 128, shows the sprocket *E* in Fig. 127 driven with the chain, keyed to a sleeve. One of the spindle change gears is attached to the other end of the sleeve, and meshes with a second change gear on a second shaft, on which a pinion of 16 teeth, cut solidly, drives a wheel of 53 teeth keyed on the spindle, so applying power close to the spindle nose. The hollow spindle and the details of its fitting will be noticed. There are a number of fixtures and attachments provided for the machine to extend its utilities by the employment of two spindles for horizontal, or face milling.

The Duplex Machine. The latest addition to the Brown & Sharpe designs is the automatic duplex machine (Fig. 129) having two headstocks, and two opposed spindles, which permits of the use of two horizontal cutters or two face mills working simultaneously. Capacities are nearly the same as those of the previous example. It takes 10 h.p. to drive at its maximum capacity, against $7\frac{1}{2}$ h.p. for the other. A single pulley is driven at a constant speed of 350 r.p.m. The table is the same size, and has the same longitudinal feed. The spindle speeds are the same, also the feeds, and their controls. The main spindle runs left hand only, the outside spindle runs in either direction, or stops. The outside head has a cross adjustment of $7\frac{5}{8}$ in.

The ratio of the time occupied in work setting to that of tooling is lessened by the utilization of the table feeds available with this machine, which are full automatic, part automatic, and intermittent. The first requires the use of a fixture at each end of the table, the work in one being tooled while the other is being loaded. The cycle is repeated automatically, the table advances at the fast rate of travel to the cutting position, cuts, reverses, advances at the fast travel to the cutting position for the fixture at the opposite end, and cuts while work is being replaced in the first. In a part automatic cycle, a single fixture being used, the work is loaded, the fast table travel engaged, cuts, and returns at fast travel and stops for reloading. Or, a swivelling fixture can be used, one station being loaded while work on the other is being cut. The intermittent cycle can be used when two or more fixtures are set in line, having varying amounts of cutting feed, and fast travel jumps between.

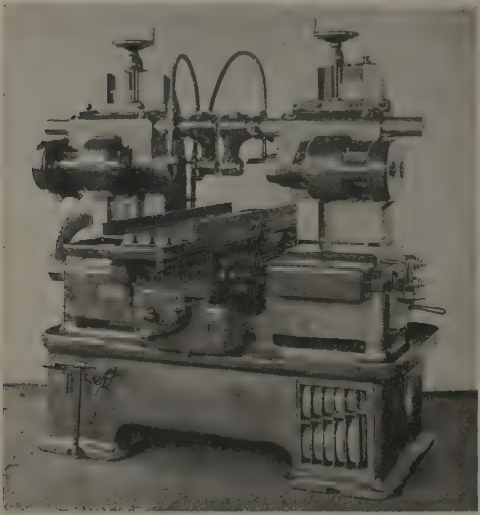


Fig. 129. Duplex-head Machine

The Pillar and Knee Design. This was the first attempt, realized by Brown & Sharpe in 1861, to include universal functions, meaning by that the cutting of spirals (helices) for which division or indexing is combined with angular settings of the work. Only that of a very light character was contemplated, and so the first machines had no overhanging arm to support the outer end of the cutter arbor, and no bracing to the knee. The headstock was simply a lathe type with a stepped belt cone, from the spindle of which the work speeds were derived also through belt cones.

In this design, the cutter arbor is continuous with the headstock spindle, and remains at a uniform height, so that vertical adjustments for depths of work have to be made with the knee. This, and the overhang of the arbor, are elements of weakness, which are corrected in all present-day machines. The arbor is supported at the outer end, and frequently at an intermediate position. The knee and arm are braced together and the knee frequently to the base. A boxed section of knee is generally substituted for the older open front design. In place of belted speeds and feeds, geared changes are now common, and feeds are independent of speeds, instead of being derived from them, with incongruous results, when using small and large cutters. And with the great increase in the weight of tables, geared feeds became imperative to avoid the slip that is liable to occur with belts. Primarily and chiefly these machines use horizontal edge cutters, but face cutters are also employed, the outer support then not being used.

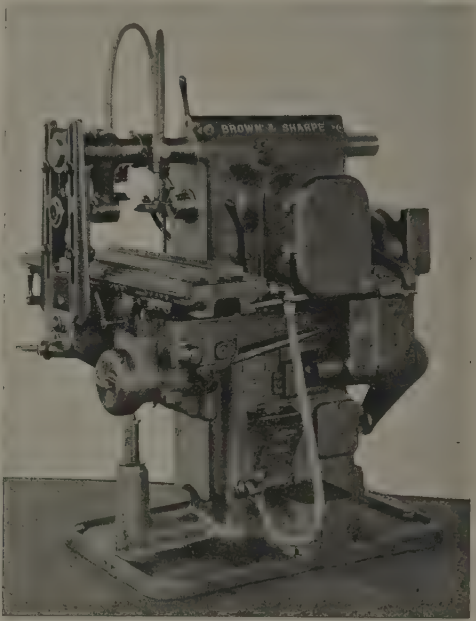


Fig. 130. Plain Miller with Automatic Control

Fig. 130 illustrates the latest design of this type of machine by Brown & Sharpe, with full automatic control, and the only machine in which the spindle is made to reverse, which permits of the use of a right- and left-hand rotary cutter on the same arbor, making possible the mounting of a fixture at either end of the table, so reducing the non-cutting time. The movements of the spindle are controlled automatically, and also those of the table, the latter including a variable cutting feed, a constant fast travel, reverse, and stop. A special feature too is the automatic lubrication. A tank is located in the base whence the lubricant is pumped to an oil reservoir at the top of the column. Thence it is supplied to all bearings within the column. A similar unit supplies the table feed. Another reservoir and pump is supplied for cutter cooling, feeding through two nozzles. In many respects the mechanism resembles that in the previous machines. A single pulley belt driven makes 300 r.p.m., and requires 5 h.p. at the maximum capacity, for which a motor can be substituted. Sixteen speed changes are provided for, from 28 to 695 r.p.m. The table has 12 speeds from 1·37 in. to 18·38 in. per minute. The range for small mills is from ·002 in. to ·050 in. per revolution of spindle, that for large mills is ·026 in. to ·656 in. per revolution of spindle.

Plain and Universal Machines. The plain machines, of which Fig. 130 is representative, are also termed manufacturing machines, because they are generally of more robust build for simple operations than the universals, and the table does not swivel. The universal differs from the other in having a saddle that swivels in a horizontal plane on a bed with veed edges, which slides on top of the knee. The table is fed with a screw of definite pitch, and carries a dividing head and a footstock. Work can be chucked on the head alone, or carried between the head, and footstocks. The head indexes and rotates the work, the swivel saddle sets it to any angle. The index plate gives circles of holes for dividing, change gears provide variable rates of spiral movements up to 45°, worm gears and change gears related to the pitch of the table feed screw give rates of rotation to the spindle and the work it carries. Heads have vertical angular movements from 10° below the horizontal, to 10° over the vertical, so providing angular settings for work in the headstock. The tailstock has a slight vertical range of angular adjustment and a vertical movement, which is used for tapered taps and reamers. By means of this mechanism, the teeth of all kinds of gear wheels can be form-cut, all kinds of milling cutters with straight or spiral teeth, and all helices up to the range of the table feed.

A good many plain machines are fitted with index centres which have no provision for setting to vertical angles, and no change gears. They have circles of holes drilled on a plate, and an index peg. They are useful for straight milling which includes a multitude of articles and tools.

The universal head in its essentials is not very different in any machines. The worm wheel has 40 teeth, and the worm is single threaded, so that 40 turns of a crank handle are required to rotate the spindle through one complete revolution. The number of turns of the crank to correspond with a fractional part of a revolution is got by dividing 40 by the number of fractional divisions desired. To turn the crank by a definite amount, the indexing plate is used which has different numbers of holes arranged in circles. A spring peg entered in a hole locks the crank. Change gears rotate the spindle slowly from the feed screw, through spiral gears, spur gears, and the worm and wheel. The ratios between the screw and the worm shaft are usually equal. Then 40 turns of the worm shaft are required for each revolution of the spindle, and the feed screw will make 40 turns. It has 4 threads per inch, so that the movement of the table will be 40 ÷ 4 = 10, termed the lead of the spiral, and the lead of the machine. Fig. 131 shows the universal head and tailstock on a Cincinnati machine, with the teeth of a spiral gear being milled with a form cutter.

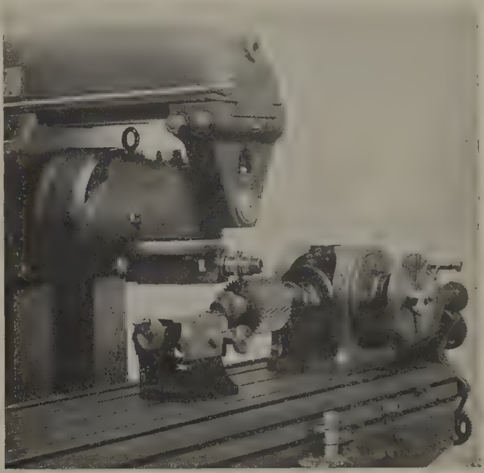


FIG. 131. CUTTING A SPIRAL GEAR

GAUGING AND INSPECTION

By Joseph G. Horner, A.M.I.Mech.E.

LESSON IV

VERNIER INSTRUMENTS

The range of utility of the vernier caliper is larger than that of the micrometer. The principle is, that a subdivision of the graduated rule is minutely subdivided by the movable vernier. The divisions vary. Generally, 1 in. on the rule is graduated into fortieths, each tenth being numbered. Twenty-four of these parts are on the vernier, subdivided into 25 parts—Fig. 23. Each space on the vernier is therefore shorter than one on the rule by $\frac{1}{25}$ of $\frac{1}{40}$, or $\frac{1}{1000}$ in. A metric vernier reads to $\frac{1}{50}$ mm.

Construction of the Caliper. One of these is shown in Fig. 24, made in sizes from 4 to 48 in. An abutment block is clamped to the beam, and a milled-head screw which it carries provides for fine adjustments of the movable jaw that carries the vernier. Though the range of vernier readings is only about $\frac{1}{2}$ in., the major graduations on the beam afford extended lengths. To read a fine dimension, since each of the 25 subdivisions on the vernier is less by $\frac{1}{1000}$ in. than the main graduations on the beam, every movement of the vernier away from the zero lines will represent as many thousandths, as $\frac{1}{1000}$, $\frac{2}{1000}$, and so on. This will increase by thousandths until the line 25 on the vernier coincides with the line 24 on the beam. To take a reading, note how many inches and tenths (0·100) and fortieths (0·025) the 0 mark on the vernier is from the 0 mark on the beam, then note the number of divisions on the vernier from 0 to a line which coincides with a line on the beam. In Fig. 25 the reading is 1 in., $\frac{4}{10}$, $\frac{1}{40}$, and $\frac{10}{1000}$, the arrow indicating the coincidence of lines on vernier and beam = 1·435 in.

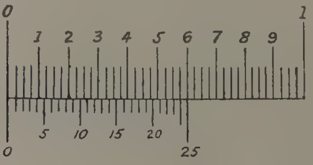


Fig. 23. The Vernier Scale

These calipers are made in several useful modifications. In some of these, the vernier is combined with the micrometer caliper to take readings in ten-thousandths of an inch. The vernier has ten divisions on the body which occupy the same space as nine divisions round the thimble. The difference between the width of a space on each is, therefore, one-tenth of a space on the thimble. The reading is identical with that just now noted. When the thimble is turned, the thousandths from zero are read, and the first line on the body that coincides with a line on the thimble. If the first line, add $\frac{1}{10000}$, if the second, $\frac{2}{10000}$, and so on.

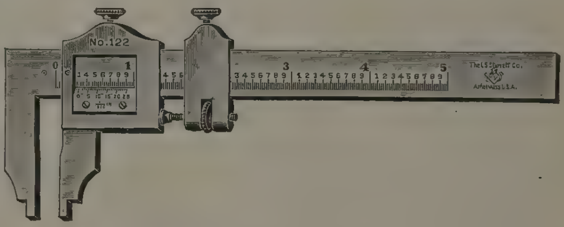


Fig. 24. Vernier Caliper

The micrometer and vernier are both applied to depth gauges for fine jig and fixture work, and both, like other models, in English and in metric measures. A vernier height gauge is shown by Fig. 26. It is used in jig and die work to ascertain the height of projections from a plane surface. The vernier reads to $\frac{1}{1000}$ in., and the stem is graduated to read from 0 to 10 in. inside of the jaws, and from 1 to 10 in. outside. The illustration shows the front side

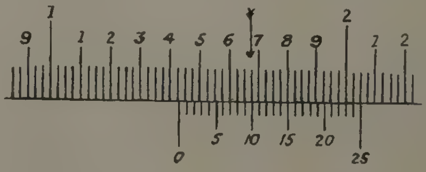


Fig. 25. Taking a Vernier Reading

of the bar (left), and the reverse side (right). On the front side when the jaws are closed, the zero lines on both stem and vernier plate will coincide, and the tool is used for outside measurements only. On the reverse side the movable jaw is drawn back to a point where the 0 lines of both stem and vernier plate coincide. Then the distance from the bottom of the base to the top of the movable jaw equals 1 in., and the instrument is ready for inside measurements only. The extension clamped to the movable jaw is a scriber which permits measurements to be taken, and lines may be drawn, or lines can be spaced and drawn parallel, and over projections in die work, and for marking out.

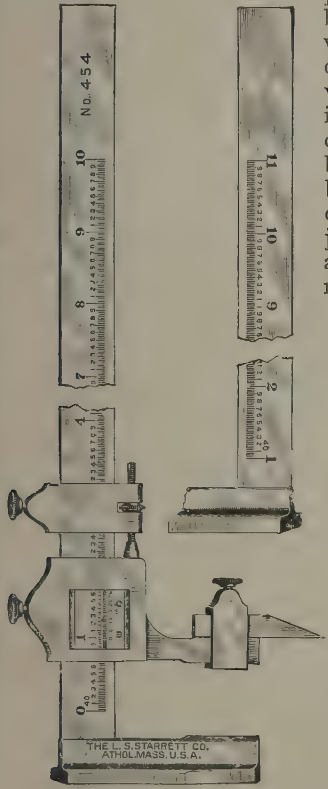


FIG. 26. VERNIER HEIGHT GAUGE

Measuring Machines. A rather large number of fine measuring machines are made, not for the common use of workmen in the shops, but in the tool room, and for the manufacture of instruments of precision. They are carried on a bed, and embody essentially the characteristic provisions of the first Whitworth machine (see page 1141), that is, an accurately cut screw of fine pitch, subdivided with a rotating graduated head, and having a gap between the end of the screw and the fixed abutment piece to receive the piece to be measured.

It may seem rather incongruous, but it is a fact that the production of an accurate screw, and the mechanism of a divided head are less troublesome than the estimation of the degree of pressure between the measuring points and the work. It occurs, though not so obvious, in all calipers, hand and micrometric. It becomes necessary in high-class measuring machines, to provide means for indicating the exact degree of contact pressure which is admissible. A common device is a feeler plug that drops out between points when a certain pressure is reached, another is a spring-operated plunger working in unison with a scale and microscope, or a graduated glass tube containing a coloured liquid, forced by the pressure of a diaphragm connected to the movable anvil, or a spirit level.

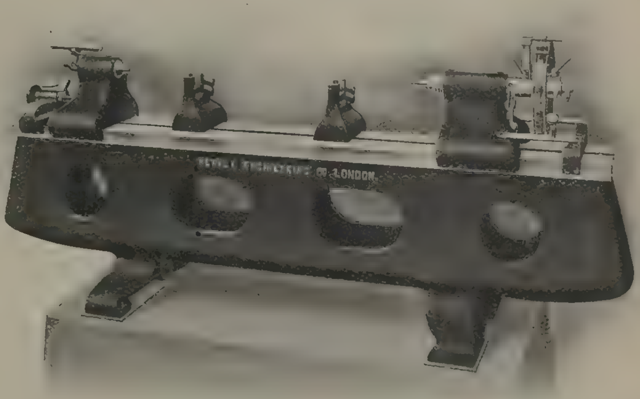


FIG. 27. NEWALL MEASURING MACHINE

The Newall machine, Fig. 27, is carried on a substantial bed, the surfaces of which are scraped and lapped to ensure alignment of the heads. The headstock gives readings to $\frac{1}{1000000}$ in., the graduations being such that the indicated size can be read directly in decimals of an inch. The headstock for metric measurements gives readings to $\frac{1}{10000}$ mm. The measuring screw has a thread of deep buttress form to give ample wearing surface. It has a range of 1 in. or of

20 mm. The measuring faces of the screw and nut are from twice to three times the length of the travel of the screw to delay wear, and preserve the accuracy of the pitch. The screw spindle runs in hardened steel bearings. An automatic adjustment maintains constant tension on the contact of the measuring screw between the effective faces of the screw and nut. Rapid movement of the screw is produced with a knurled nut at the end of the spindle, which is used until sufficient pressure has been applied through the piece being measured to bring an indicator into motion. A fine adjustment screw gives a sensitive movement to the measuring screw.

The pitch of the measuring screw is subdivided by the graduations round the measuring wheel seen at the right of Figs. 27 and 28, in combination with a scale and vernier, divided decimally. In Fig. 29 the method of reading is shown, giving 0·31254 in. obtained thus: the first digit 3 is the highest figure disclosed on the left-hand side of the scale that carries the vernier. The second and third digits, 1 and 2 respectively, appear as the highest main graduations opposite the vernier, the fourth digit 5 is the highest subdivision on the measuring wheel below the zero line on the vernier, and the fifth digit 4 is the graduation on the vernier that coincides with any graduation on the measuring wheel. As the pitch of the measuring screw for English dimensions is 20 threads per inch, it is necessary to add 0·05000 in. to the indicated size in all cases where the vernier may have passed the subdivision between any two main divisions on the scale that carries the vernier. Thus, in Fig. 29, if the measuring wheel had been given one complete revolution outwards, the subdivision between digits 3 and 4 on the scale would be disclosed, and the reading would be: 0·31254 + 0·05000 = 0·36254 in.

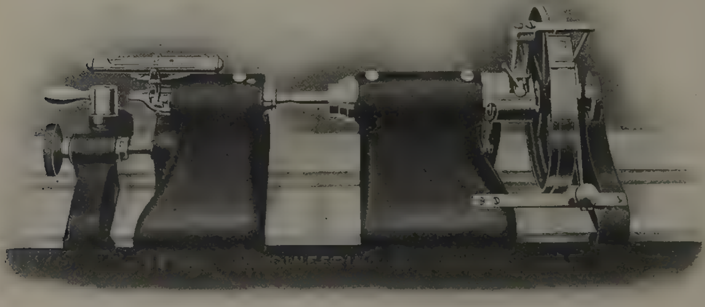


Fig. 28. Measuring Machine Set to Zero

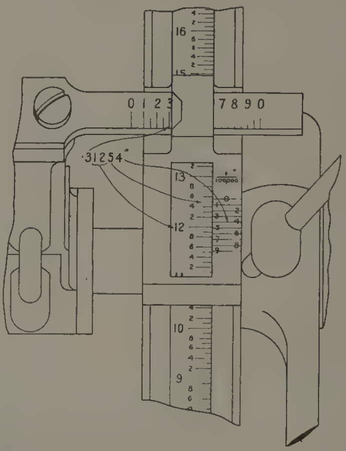


Fig. 29. How to Read the Measuring Machine

The tailstock, seen at the left of Figs. 27 and 28, has an auxiliary saddle behind it, with a fine screw connection to the tailstock to provide a very delicate movement for setting, either to zero, or to graduated lines on a rule. The mechanism for setting and testing is extremely delicate. For measurements up to 1 in. the anvil faces of head and tail stocks are brought into close proximity, about $\frac{1}{32}$ in. apart, with the reading on the vernier approximately zero. But for dimensions over 1 in., end measuring rods are supplied, or rules and a microscope. For lengths above 1 in. the length of measuring rod employed must be added to the reading obtained from the headstock. An indicator and a spirit-level on the tailstock, Fig. 28, show the

(*Continued on page* 1160)

STEAM BOILERS AND THEIR ACCESSORIES

By Percy E. Rycroft, M.B.E., M.I.E.E.

LESSON IX

AIR HEATERS

The utilization of preheated air in the boiler furnace as a means of still further improving combustion efficiency has considerably extended during recent years.

Even in a well-arranged, steam-raising plant, comprising boiler, superheater, economizer, and mechanical stoker, the temperature of the furnace gases entering the chimney base is rarely much below 400° F.

It is obvious, therefore, that if a still further portion of this otherwise waste heat can be reclaimed, an improved efficiency must result.

As a rough basis of calculation, it may be assumed that the overall efficiency of a steam plant can be increased by about 3 per cent if air heaters are added to a plant in *addition* to economizers.

Where air heaters are installed as a *substitute* for economizers, the prevailing conditions are usually so varied that no definite comparison can be laid down, and each case has to be determined on its merits.

In many modern power installations for instance, it is now customary to "bleed" the turbines at one or more of the steam stages and utilize this steam for raising the temperature of the boiler feed water. This being so, it will be seen that as the boiler feed under such conditions can be supplied direct to the boiler at or near boiling point a very large economizer heating surface would be required for any appreciable amount of additional heat to be imparted from the flue gases. Under such conditions the benefit to be derived from an economizer would probably not warrant the capital expenditure involved, and an air heater might be installed in its place.

On the other hand, the installation of an air heater involves the utilization of forced draught

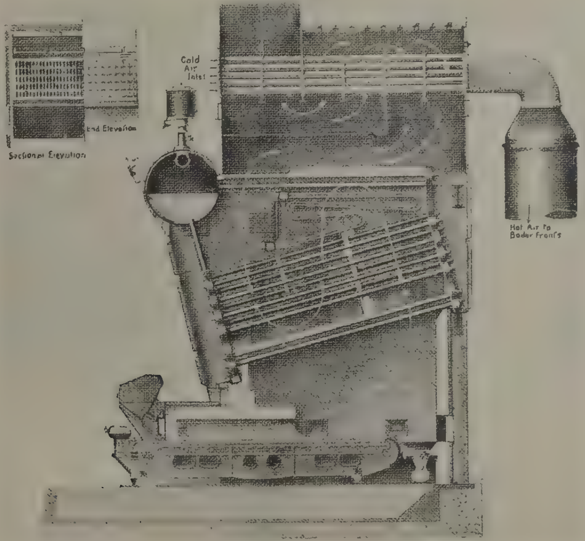


Fig. 56. Green's Rectangular Tube Air Heater Over a Water-tube Boiler

at all times if the heated air is to be effectively used, and the question of the amount of power absorbed by fans and other auxiliaries in this connection has, therefore, to be taken into consideration when comparing relative costs. This would apply more especially where the installation is such that during certain periods of the day—but for the necessity of utilizing the hot air—it could normally be run quite efficiently under natural draught conditions.

It is particularly important to remember that the modern steam-raising plant has to be considered as a whole, and that no one part, such as an economizer, superheater, or air heater, can be taken away from or added to an existing installation without entailing some modification to the other parts if a maximum efficiency is to be retained. If, for instance, an air heater is introduced into a design, then the economizer heating surface will probably be less than if an economizer is used alone. If, on the other hand, an air heater only is installed without an economizer, then the boiler heating surface may be increased for a given capacity, and so on.

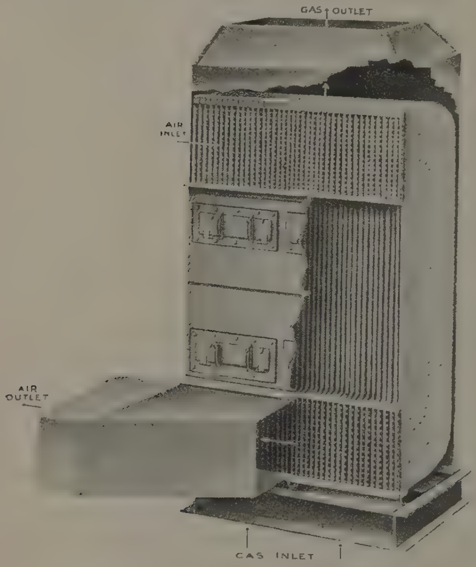


Fig. 57. The "Usco" Air Heater

The effect of heated air on mechanical stokers and furnace linings limits the degree of temperature which can be usefully and commercially employed. Normally, 400° F. should be considered as a maximum air temperature, while, for general running, about 250° F. is ample.

The general function of an air heater is to transfer heat units from the flue gases to the incoming furnace air.

The apparatus employed takes several forms, such as (*a*) a tubular construction, in which the flue gases pass through a number of tubes while the incoming air passes across the outside of them; (*b*) cellular plate construction, in which the flue gases pass between alternate pairs of flat plates spaced a slight distance apart while the incoming air passes on either side; and (*c*) direct contact heaters in which groups of thin corrugated plates, by continuous mechanical movement, are alternately made to pass through the flow of the furnace gases and the incoming air, so that the heat absorbed from the former is given up to the latter.

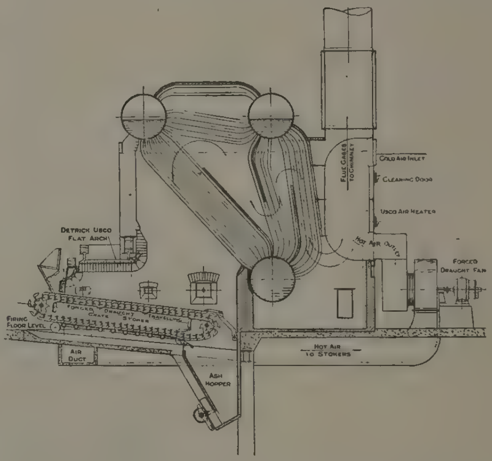


Fig. 58. Arrangement of "Usco" Air Heater

Fitted in the space between the rear wall of the boiler and the boiler tubes, a compact setting whereby radiation losses are minimized

Tubular Air Heaters. The tubular air preheater—developed by Babcock & Wilcox, Ltd., and also by Messrs. Yarrow—is of simple construction, consisting of a number of light gauge (No. 10 or less) tubes usually about 2¾ in. in diameter, expanded at either end into steel plates, the whole tube nest being enclosed in steel casing. The flue gases pass through the tubes on their way to the chimney, while the incoming furnace air passes over the outside of them.

Baffles are arranged within the casings so that the incoming air makes two or more passes across the tubes before entering the furnace air ducts. Any soot deposit is readily removed by blowing superheated steam through the tubes from time to time.

The Green's tubular air heater, Fig. 56, is of rectangular section, and is constructed of cast iron. It is built up from elements consisting of four rectangular tubes cast in one piece with connecting flanges at either end. The tubes are 2 ft. 6 in. long by 4 in. deep and 1 in. wide, and are spaced $1\frac{1}{8}$ in. apart.

The incoming air passes through the tubes while the furnace gases transversely sweep across the outside. Both the gases and the air can be made to pass either straight through the heater or through sections of it in series, according to the volume which has to be dealt with and the degree of temperature required.

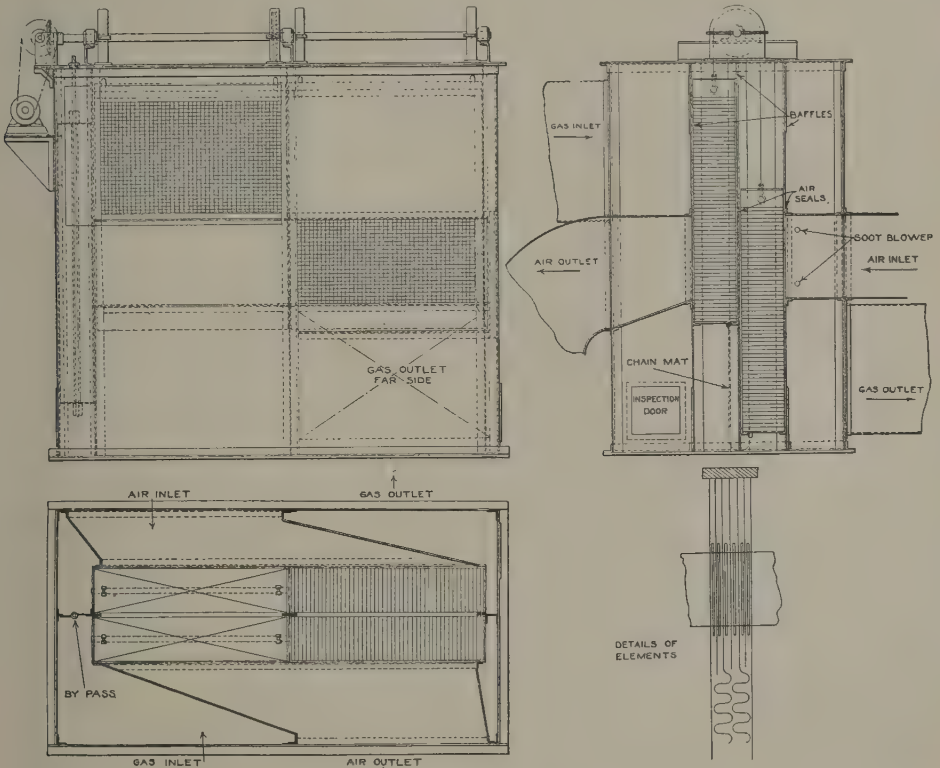


Fig. 59. The Perry Air Preheater

The Usco Air Heater. The "Usco" apparatus is of the plate type, built up from thin metallic sheets by means of which the gas and air streams are divided up into very thin alternate films separated by the very thin metal plates. The gas and air inlets are arranged on the contra-flow principle, the air to be heated meeting continually hotter surfaces as it passes through the apparatus. The gas passes through the heater in a perfectly straight path, while the air to be heated enters and leaves by an easy curve. The internal resistance of the apparatus is low and, to avoid all possibility of leakage between the air and gas passages, all joints used in the construction of an element are welded. The general construction of the heater will be apparent from Figs. 57 and 58. All surfaces are readily accessible for cleaning either by means of a brush or steam operated lance.

The Howden Ljungström. The Howden Ljungström air preheater has a number of special features, and is a self-contained unit of particularly small dimensions.

It consists of a main rotor carrying heat transferring elements enclosed in steel casings, with which are combined induced and forced draught fans.

The flue gases enter at the underside of a cylindrical steel casing, pass through the rotating elements, and discharge to the chimney above.

The cylindrical casing is divided vertically by a central partition, so arranged above and below the rotor as to prevent leakage of the flue gases into the incoming air.

The cold air enters at the upper end of the cylindrical casing, and passes downwards through the rotor into the furnace air ducts below.

The induced draught fan on the chimney side and the forced draught fan on that of the incoming air are driven from a common shaft, and it will be noted that the flue gas and the air pass through the rotating elements in opposite directions, so that a contra-flow effect and high efficiency of heat transmission is obtained.

The rotor, which is built up of welded steel plates, revolves at a speed of about 6 rev. per minute, and carries the heat transferring elements.

These are made up from corrugated steel sheets $\frac{1}{50}$ in. thick, placed between flat sheets of similar thickness packed together in such a way as to form a large number of small vertical and triangular passages, and provide a large and effective heating surface.

The Perry Air Heater. The Perry air heater is also arranged on the principle of direct contact, but differs from the preceding apparatus in the fact that a reciprocating action is adopted instead of a rotary one, and that two separate elements are employed.

The elements are built up from very thin corrugated brass plates, and are suspended from a chain passing over a pulley. Motion is imparted to this pulley by an electric motor, and an automatic reversing mechanism comes into action when the elements reach the limit of their upward or downward travel.

The elements pass alternately through the path of the flue gases and that of the incoming air, and a high rate of heat transference is claimed. Special precautions are taken against the infiltration of gas into the air space and vice versa by the provision of metallic brushes between the elements and the openings in the casings through which they pass.

From the illustration, it will be noted that the two elements are enclosed in a steel casing, through which the air duct passes and round which the hot gases circulate.

Where the flue gases enter they divide into two streams, one of which passes through the upper element and the other through the lower, both streams uniting again at the outlet.

The air to be heated passes through the heater on the contra-flow principle and, as will be seen, must always pass through both elements on its way to the furnace.

GAUGING AND INSPECTION

(*Continued from page* 1156)

degree of pressure on the anvil by a movement of the bubble, which magnifies that of the anvil about 4,000 times. The possibility of varied readings by different operators is eliminated, and changes due to temperatures are seen. The vee rests seen in Fig. 27 support end measuring rods. The machine is used for measuring and checking gauges, to detect alterations due to wear, for duplicating standards, and the possibility of personal error is practically eliminated.

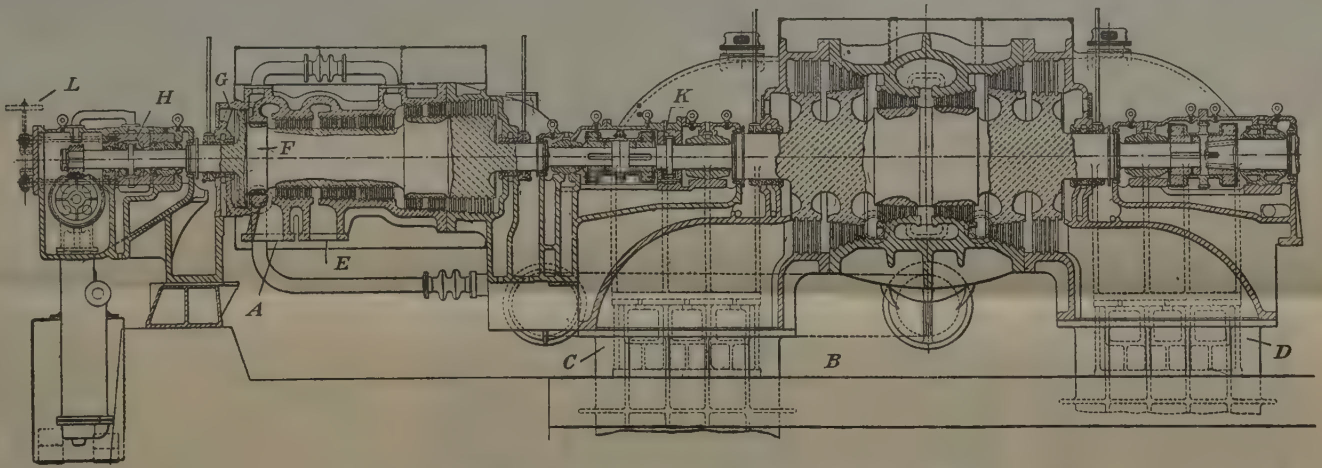


Fig. 38. Longitudinal Section Through Parsons Tandem Turbine

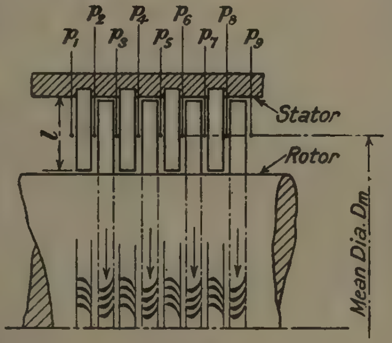


Fig. 39

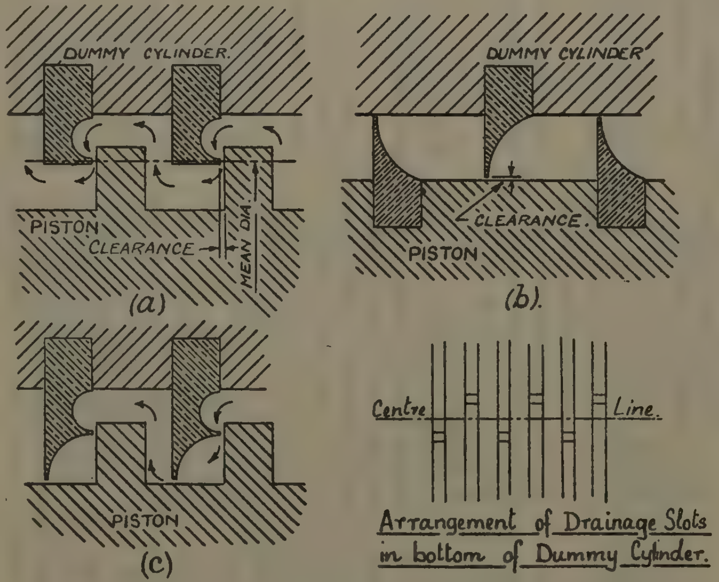


Fig. 40. Details of Dummy Pistons for Parsons Turbines

STEAM TURBINES

By W. J. Kearton, M.Eng., A.M.I.Mech.E., A.M.Inst.N.A.

LESSON IX

REACTION STEAM TURBINE CONSTRUCTION

Parsons Turbine

General Description. Fig. 38 shows a longitudinal section through a modern two-cylinder tandem Parsons turbine suitable for high powers. The high pressure unit is a single-flow turbine, and may be taken as an example of the normal Parsons turbine for low and moderate powers, with the exception that the number of stages in a single cylinder high pressure turbine would be considerably greater than that in the high pressure unit of the tandem machine. The low pressure unit is a double-flow machine.

Live steam, after passing through the governor valve, enters the steam belt by the branch *A*. It flows to the right through the complete blade system of the high pressure turbine, and enters the low pressure turbine by the receiver connections *B*, one of these being placed at each side of the turbine. On entering the low pressure turbine the steam divides into two approximately equal amounts, one half flowing to the left, and the other half flowing to the right, the exhaust steam then entering the underslung condenser by way of the exhaust branches *C* and *D*. By the use of a double-flow low pressure turbine, the weight of steam passing through the turbine per second may be doubled. Hence, the maximum power that may be developed by the machine is twice that of a single-flow machine operating under the same conditions as regards peripheral velocity, rotational speed, and vacuum.

It is usual in Parsons turbines, to divide the whole of the blading into groups, or "expansions," as they are called, of equal height. In the small diameter high pressure sections, an expansion may comprise several rows of blades, while in the large diameter low pressure sections an expansion may only have one ring of fixed and one ring of moving blades. In the example shown in Fig. 38, there are seven expansions in the high pressure turbine, and ten in the low pressure turbine.

The high pressure turbine is made relatively small in diameter so as to diminish the loss by leakage caused by the pressure difference across each ring of blades, whether fixed or moving, and to minimize the risk of distortion due to temperature variation. In order to withstand the high temperatures satisfactorily (for reasons which have been explained in a previous lesson), the steam end of the high pressure turbine is constructed of cast steel. The exhaust end of the high pressure turbine and the whole of the low pressure turbine casing are of cast iron.

An alternative inlet branch for the steam is shown at *E*. This is used for overloads, the steam then flowing directly into the second expansion of the blading. As the total area through the channels of the blades in the second expansion is greater than that through the blades of the first expansion, the total weight of steam which may flow through the turbine is automatically increased, and hence the total power developed is greater.

The Rotor. In Parsons turbines, in which the peripheral velocity of the blades is relatively small, the rotor is frequently a forged steel drum shrunk on spiders or solid forged steel ends. In rotors of relatively small diameter and with higher peripheral speeds, the rotor is a solid forging as shown in Fig. 38. In low pressure rotors the solid rotor is lightened somewhat by turning off material as shown, so leaving heavy discs of great strength.

In true impulse turbines the pressure in the blades is sensibly constant, and the only thrust on the rotor is that due to friction in the blade channels. This is usually of negligible amount. In reaction turbines, however, there is a considerable thrust due to steam pressure. Consider Fig. 39, which represents, in diagrammatic form, an expansion of a reaction turbine. With "half-degree" reaction, the pressure drops $p_1 - p_2$, $p_2 - p_3$, $p_3 - p_4$, . . . $p_8 - p_9$ will be approximately constant, provided the total pressure drop $p_1 - p_9$ is not very great.

Let A = annular area of one blade ring, in sq. in.
Let l = radial height of blades (in.).
,, D_m = mean diameter of blade ring (in.).
Then $A = \pi . D_m . l$. sq. in.

It follows that the axial force on each ring of moving blades will be equal to $(p_2 - p_3)\,A$, and the total thrust will be

$$4\,(p_2 - p_3)\,A$$
$$\text{or } \tfrac{1}{2}\,(p_1 - p_9)\,A.$$

In the turbine shown in Fig. 38, the axial thrust on the blades of the first three expansions of the high pressure turbine is more or less balanced by the "dummy piston" *F*, which has a diameter about equal to the mean diameter of these expansions. A second dummy piston *G* balances the thrust on the blades of the last four expansions. The steam pressures on the balance pistons are made equal to those at outlet from the third and seventh expansions respectively, by means of the two pressure-equalizing pipes which are clearly shown in the illustration.

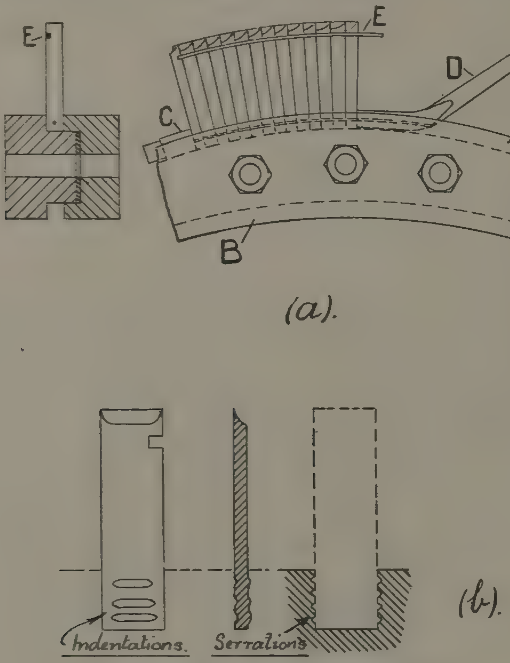


Fig. 41. Turbine Blade Former

The steam thrusts on the two sets of low pressure blading are, of course, mutually balanced, and no dummy piston is necessary.

Leakage of steam past the dummy pistons is minimized by the use of labyrinth packing. Three designs of such packing are illustrated in Fig. 40. The action of labyrinth packing has already been explained, and here we need only concern ourselves with details of construction. The packing shown in Fig. 40 (*a*) is commonly used at the high pressure end of land and marine turbines. Rectangular grooves are turned in both the rotor and cylinder. L-shaped packing strips are caulked into the cylinder grooves. The working faces of the packing strips are then ground up at a slow speed, after which the turbine is warmed up and the rotor set back to give a small clearance of from ·01 to ·02 in.

In low pressure astern marine turbines the differential expansion of the rotor and cylinder is too great to permit of the use of this kind of packing, and the "radial" packing shown in Fig. 40 (*b*) is adopted. Here the clearance is a radial one and cannot be adjusted. An improved form of packing is shown in Fig. 40 (*c*). This is a combined "facial" and "radial" packing and gives two throttlings for each packing strip.

Drainage slots are provided in the packing strips at the bottom of the dummy cylinder.

These are staggered, as shown, so as to baffle leaking steam.

In the turbine illustrated in Fig. 38, the axial position of the turbine rotor is determined by the single-collar pivoted-pad type adjusting block. The latter may be moved axially by two screwed rods which may be rotated through a worm gear by the hand-wheel *L*. The axial position of the low pressure rotor is fixed by the block *K*. Since the introduction of the Michell block, dummy balance pistons have frequently been omitted from marine geared turbines.

Blades. The blades of most Parsons turbines are of brass, rolled to standard sections, to suit standard widths of grooves. At the high pressure end of the turbine, steel blades are used for superheated steam. Comparatively little machining is required to finish the blade ready for insertion in the rotor or cylinder.

Various methods have been proposed and used from time to time, but those usually adopted in practice divide themselves into two classes, viz., the "segmental" and "individual" methods.

In the segmental method, the blades are fitted together in a jig, or "former," so as to form a segment of the complete blade ring prior to being caulked into the cylinder or rotor. The "former" consists of a cast-iron segment in which two grooves are turned of the same radius and depth as those in the cylinder and rotor respectively. A typical former is shown in Fig. 41 (*a*). *A* is the rotor groove and *B* the cylinder groove. Each groove is about 0·005 in. less in width than the corresponding rotor or cylinder grooves.

The blades are separated by standard distance pieces, and a segment is built up by threading distance pieces and blades alternately on a root wire and frequently caulking by means of a curved set *D*. When the complete segment has been formed, the blades are set radial by means of a square, and the ends bound together by means of a binding strip *E* which is inserted in a small recess machined at the inlet side of the blade and is silver soldered to the blades. When the segment is finished, it may be transferred to either the turbine cylinder or rotor to which it is attached by radial caulking of the distance pieces.

In the "individual" system, illustrated in Fig. 41 (*b*), the root wire is dispensed with and, as the name implies, each blade is inserted separately. Each blade is indented at the root, in a press. Serrations are turned in the sides of the grooves, and the principle of the attachment is that by the caulking of the distance pieces, the blade is firmly held by the distance pieces and they, in turn, are firmly held by the serrations. With ordinarily good workmanship this method of blading produces a strong and reliable attachment.

One of the greatest sources of loss in Parsons turbines is the steam leakage across the tips of both the fixed and moving blades. This is especially the case at the high pressure end of the turbine, owing to the short blades, the relatively greater clearance, and the larger pressure drops. In recent years, a form of blading, known as "end-tightened" blading, has been introduced. This is illustrated in Fig. 42. Described briefly, a shrouding strip *C* attached to the blade tips runs close to the root *B* of the next adjoining blade. The clearance which controls the leakage of steam is, therefore, an axial one, and may be adjusted to a very small amount.

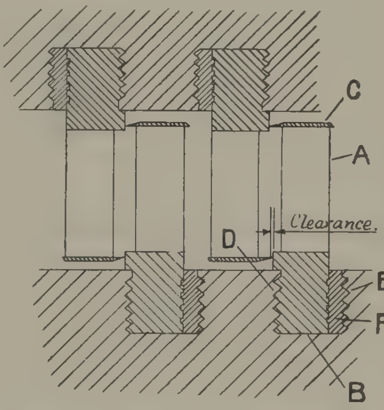


Fig. 42. Parsons "End-tightened" Blading

In very large turbines recently constructed by Messrs. C. A. Parsons & Co., steel blades are used in which the blade and root are formed integrally by *rolling*, a new process devised by Messrs. Parsons.

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XXIII

Ordering the Part. Proceeding with the drawing office method of dealing with an inquiry for a ship's propeller, we may assume that the order has been obtained. We now have to set the machinery in motion for manufacturing the part. This is done, apart from the anticipatory information to the shops from the management, by issuing a working drawing of the propeller to the shops.

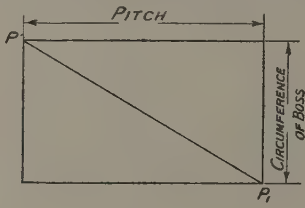


Fig. 1A

The chief draughtsman, in giving instructions for the working drawing to be made, will not necessarily give the work to the draughtsman who made the tender drawing. He will consider it sufficient to give a draughtsman the tender drawing, which will now be considered the *basis*, or *design* plan, together with the owners' specification.

In commencing the working drawing, with the above tender drawing as the chief guidance, we may assume this plan to be satisfactory with regard to both strength and form. We may now give our attention solely to any practical points which may arise while the drawing is in progress.

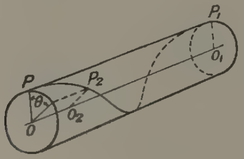


Fig. 1B

Referring to other working drawings of propellers for general guidance, we may observe that the usual method of projecting points on the surface of the blade is quite approximate. Consequently, the projected surface of the blade produced from sections determined by the above method will be approximate.

Considering the empirical nature of the rules used in finding suitable blade surfaces, the error is not serious, and is quite justified by the simplification given to the representation. There is, however, one section of the blade which requires a more careful determination, i.e. the section at the juncture of boss and blade. We must provide a surface of landing, on the boss, sufficiently large to take the blade.

For the above reason it is advisable to adopt the more accurate method of projection for the section at the boss in fixing the length of same.

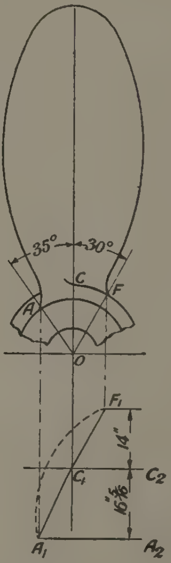


Fig. 2

Correct Projection. Referring to Fig. 1A, we have a rectangle having a length equal to the propeller pitch, and a width equal to the circumference of the boss. A diagonal is drawn from P to P_1, and the surface rolled into a cylinder as shown in Fig. 1B. The line PP_1, has now formed a helix on the cylinder. If we imagine the line OP to rotate in a clockwise direction as it moves towards the line O_1P_1, the rotary and linear movements being uniform throughout the travel, we may obtain a simple relationship between the angular and linear positions of the moving line relative to O_1P_1.

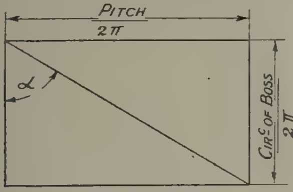


FIG. 3A

Thus, for any intermediate position on the helix P we have

$$\theta^\circ : O.\ O_2 :: 360^\circ : O.\ O_1.$$

$$\theta^\circ : O.\ O_2 :: 360^\circ : \text{pitch}.$$

$$O.\ O_2 = \frac{\theta^\circ}{360} \times \text{pitch}.$$

Referring to Fig. 2, the point A represents the point of intersection of the aft edge of the blade, where it joins the boss, which at this point has a diameter of 26 in. The distance between the planes A_1A_2 and C_1C_2 being the linear movement of the point on the helix in revolving through the angle AOC and will equal $\frac{35}{360} \times 168 = 16\frac{5}{16}$ in.

The line AA_1 will support the point representing the aft edge of the blade adjoining the boss. The line OC_1 will represent the axis of rotation and the line C_1C_2 a plane normal to the axis OC_1, the point C_1 representing the place where the leading edge of the blade reaches the summit of the boss.

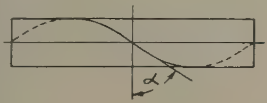


FIG. 3B

The point F_1 having been obtained in a manner similar to the above, a curve may be drawn through the points $A_1C_1F_1$, representing a plan view of the leading edge of the blade.

A comparison of the above curve $A_1C_1F_1$ with the tender drawing (Fig. 5, on page 1072) will show that, to ensure a landing for the blade, it will be necessary to move the blade nearer to the forward face of the boss.

Approximate Projection. We may now, for the purpose of comparison, draw a plan of the blade at the boss using the approximate method. Referring to Fig. 4, if we now draw a rectangle to the same proportion as the above, but to a reduced linear scale of $1 = 2\pi$, the width will equal the radius of the boss and the length will equal the pitch $\div 2\pi$, as shown in Fig. 3A. Fig. 3B represents the above rectangle rolled into a cylinder with the angle α representing the angle made by a tangent to the helix, at the crown of the boss, to the plane of revolution.

Referring to Fig. 4, OC_1 represents the axis of rotation, and the lines OF_1 and C_1C_2 planes of rotation separated by a distance equal to the pitch $\div 2\pi$. A line drawn from F_1 to C_1 will be tangential to the helix at C_1, but not at the point F_2. Similarly, the line A_1C_1 will be tangential to the helix, on a cylinder having a diameter equal to the diameter of boss at the aft end, at the point C_1 but not at the point A_2.

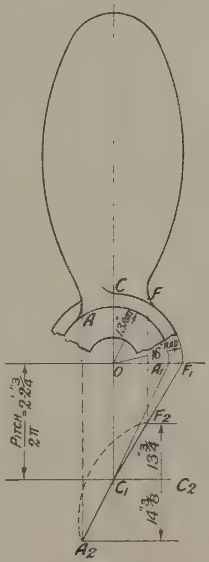


FIG. 4

If we assume the line drawn through the

points $A_2C_1F_2$ represents the leading edge of the blade adjoining the boss, we fail to obtain a true length of blade section, and it is this approximation which was made in producing the tender drawing.

Having, from Fig. 2, determined the correct position for the blade on the boss, we may proceed with the working drawing, applying the above approximate method to determine the lengths of blade sections at the various radii and for finding the outline of the developed blade.

Construction Lines on Working Drawing. We have, in a previous lesson, referred to the lines of construction used in the making of a working drawing, which unless made on a sheet of tracing paper above the drawing should be erased. In the working drawing of the above propeller (Fig. 5) it will be observed that certain lines of construction are left on the drawing. This is done for the information of the pattern-maker, and provides an example of a drawing in which the lines of construction may be of service to the craftsman in making the part.

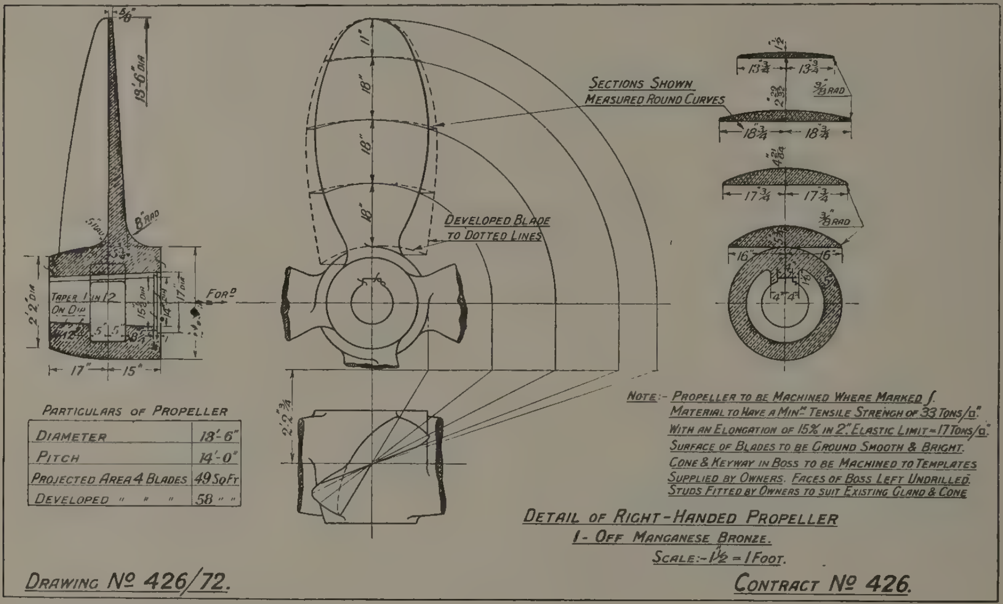

5/8"
19'-6" DIA
8" RAD
5/16 RAD
2'2" DIA
TAPER 1 IN 12 ON DIA
15 3/8 DIA
14 DIA
17" DIA
FORD
17"
15"
PARTICULARS OF PROPELLER
DIAMETER 18'-6"
PITCH 14'-0"
PROJECTED AREA 4 BLADES 49 SQ FT
DEVELOPED " " " 58 " "
11"
18"
18"
18"
DEVELOPED BLADE TO DOTTED LINES
SECTIONS SHOWN MEASURED ROUND CURVES
13 3/4
13 3/4
3/8 RAD
18 3/4
18 3/4
17 3/4
17 3/4
3/8 RAD
16
16
4
4
2'2 3/4"
NOTE:- PROPELLER TO BE MACHINED WHERE MARKED ∫.
MATERIAL TO HAVE A MINM TENSILE STRENGH OF 33 TONS/□".
WITH AN ELONGATION OF 15% IN 2". ELASTIC LIMIT = 17 TONS/□".
SURFACE OF BLADES TO BE GROUND SMOOTH & BRIGHT.
CONE & KEYWAY IN BOSS TO BE MACHINED TO TEMPLATES
SUPPLIED BY OWNERS. FACES OF BOSS LEFT UNDRILLED.
STUDS FITTED BY OWNERS TO SUIT EXISTING GLAND & CONE
DETAIL OF RIGHT-HANDED PROPELLER
1- OFF MANGANESE BRONZE.
SCALE:- 1½" = 1 FOOT.
DRAWING Nº 426/72.
CONTRACT Nº 426.

FIG. 5

EXERCISE

Complete the exercise of the last two lessons by making a working drawing of the propeller in question.

MATHEMATICS FOR ENGINEERS

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LESSON XXIV

DIFFERENTIATION—(*contd.*)

Small Quantities. Another use of differentiation is to enable us to calculate small variations in one quantity consequent upon small variations of a related quantity. For instance, a fly-wheel is used to absorb the fluctuations of energy, and must be designed so that this fluctuation of energy entails only a small variation of speed. If we can find a connection between the variation of energy, δE, and the consequent variation of speed, δn r.p.m., we shall be able to choose the weight (or, more accurately, the moment of inertia) so that the value of δn may be kept within prescribed limits.

To obtain a formula covering all such cases, we remember that the differential coefficient was defined as the limit to which the ratio of the small increments tended as the increments themselves tend to zero. Thus, provided that δx is small, $\frac{\delta y}{\delta x}$ and $\frac{dy}{dx}$ must be nearly equal, i.e.,

$$\frac{\delta y}{\delta x} \doteq \frac{dy}{dx} \text{ or } \delta y \doteq \frac{dy}{dx}\,\delta x.$$

It must be noted that this is an *approximation* formula, and the approximation is good so long as δx is so small that the variation of $\frac{dy}{dx}$ in the interval δx can be neglected.

Example. Find the weight of a fly-wheel, the radius of gyration being 2 ft., to keep the variation of speed of a double-acting engine developing 50 h.p. at 120 r.p.m. within 1 r.p.m. on each side of the mean, the fluctuation of energy being 10 per cent of that of one stroke.

Here we have, denoting the energy of the fly-wheel by E and the speed by n (r.p.m.),

$$\delta E \doteq \frac{dE}{dn}\,\delta n.$$

Let w be the weight of the fly-wheel, when the kinetic energy will be $\frac{w \times 4}{2 \times 32{\cdot}2}\left(\frac{2\pi n}{60}\right)^2 = 0{\cdot}000680 w\, n^2$ ft.-lb. $= E$. So $\frac{dE}{dn} = 0{\cdot}00136 wn$, and since $\delta n = 2$,

$$\delta E \doteq \frac{dE}{dn}\,\delta n = 0{\cdot}00136 w \times 120 \times 2 = 0{\cdot}3264 w.$$

But δE is also 10 per cent of the energy developed per stroke, i.e.

$$\delta E = \frac{10}{100} \times \frac{50 \times 33{,}000}{240} = 687{\cdot}5 \text{ ft.-lb.}$$

Therefore, $\quad 0{\cdot}3264 w = 687{\cdot}5$

$$w = \frac{687{\cdot}5}{0{\cdot}3264} = 2{,}100 \text{ lb., nearly.}$$

Another use of the approximation formula is to obtain numerical approximations. We will show how to use calculus to obtain a good approximation to $\sqrt{26}$. We know the answer is just over 5, since $\sqrt{25} = 5$. Thus, if $y = \sqrt{x}$, we know that $y = 5$ when $x = 25$, and want y when $x = 26$; we proceed to find the alteration in y when x increases by 1, i.e. the value of δy when $\delta x = 1$.

Now, $\quad \delta y \doteq \frac{dy}{dx}\,\delta x$

and, since $y = \sqrt{x} = x^{\frac{1}{2}}$

$$\frac{dy}{dx} = \tfrac{1}{2}x^{-\frac{1}{2}} = \frac{1}{2\sqrt{x}} = \frac{1}{2 \times 5}$$

when $x = 25$.

Therefore, $\quad \delta y = \frac{1}{2 \times 5} \times 1 = 0{\cdot}1$

Therefore an approximation to $\sqrt{26}$ is $5 + \delta y = 5{\cdot}1$. (Tables give $5{\cdot}0990$ . . .)

Approximate Solution of Equations. In the above, we have, in effect, obtained an approximation to the solution of the equation $y^2 = 26$, starting from the value 5 which is a rough approximation. A similar method can be used to improve a rough approximation to the solution of *any* equation, and by successive applications of the method, we may obtain the root to *any* required degree of accuracy. We will apply the method to the solution of the equation $x^3 - 9x + 2 = 0$, which we solved by graphical methods in Lesson X.

By trial, we soon find that $x = 3$ is a rough approximation to the root. Denoting $x^3 - 9x + 2$ by y, we have

When $x = 3, y = 3^3 - 9 \times 3 + 2 = 2.$

We want $y = 0$, so we have $\delta y = -2$; and must find the corresponding value of δx. To do this, we use the formula

$$\delta y \doteqdot \frac{dy}{dx}\,\delta x.$$

Since $$y = x^3 - 9x + 2,$$

$$\frac{dy}{dx} = 3x^2 - 9 = 27 - 9 = 18$$

when $x = 3$. Substituting the values of δy and δx, $-2 = 18\,\delta x$, so $\delta x = -\frac{1}{9} = -0{\cdot}111\ .\ .\ .$

Thus, a better approximation to the root is $3 + \delta x = 3 - 0{\cdot}111\ .\ . = 2{\cdot}889$.

Should we need a still better approximation, we use the same process, starting with 2·89, say, instead of 3, and by successive approximation we may obtain our root correct to as many places of decimals as we need.

EXERCISE No. 37

1. Find the weight of the fly-wheel of a small engine developing 5 h.p. at 1,200 r.p.m., to limit the total fluctuation of speed to 5 per cent of the mean, the fluctuation of energy being 20 per cent of that per revolution. Take the radius of gyration as 6 in.

2. Use calculus to find approximations to $\sqrt{84}$, $\sqrt{45}$, and $\sqrt[3]{60}$.

3. Obtain the root of the equation $x^3 + 10x = 31$ near 2 correct to three places of decimals.

INVERSE RATE OF CHANGE

In a very large number of practical cases, we are able to calculate the rate of change of a quantity by means other than differentiation, but need then a formula for the quantity itself. For instance, in mechanics, we can calculate the accelerations of moving bodies from their weights and the forces acting on them, and need their velocities and displacements.

Suppose we are told that

$$\frac{dy}{dx} = 6x$$

what must be the value of y? We must remember the rule for differentiation and try to construct a formula for y such that when it is differentiated, we obtain $6x$. Now in differentiating, we reduce the index by 1, so that to get x we must have started with x^2. Also in differentiating, we multiply by the index, in this case 2, so that we must have had for the coefficient of x^2, $6 \div 2 = 3$. Thus, $y = 3x^2$ is a possible value of y. But is it the *only* possible value? Well, if we put $y = 3x^2 + 2$, we still have $\frac{dy}{dx} = 6x$. And the same is true if we have *any* constant instead of 2, since the differential coefficient of *any* constant is zero. Thus the value of y is

$$y = 3x^2 + C$$

where C may be any constant, positive, or negative. In the same way, if $\frac{dy}{dx} = 15x^2$, we have $y = 5x^3 + C$, and so on. The rule must be clear by now; it is the reverse of the rule for differentiation,

Add 1 to the index.
Divide by the new index.
Add a constant.

The presence of an unknown constant may cause some misgiving, making it appear as though the result is not certain and determinate. We must remember, however, that in problems of this type which occur in practice, there will always be additional information, and we shall see that this always enables us to find the appropriate value of C for the problem in hand. For instance, if we know that a body has been moving for 3 sec. with an acceleration of 4 ft./sec.², we know that its velocity has increased by $3 \times 4 = 12$ ft./sec.—but unless we know its initial velocity, we cannot find its final. Here

$$\frac{dv}{dt} = 4$$

therefore $$v = 4t + C$$
$$= 12 + C \text{ when } t = 3$$

and we see that in this case C stands for the initial velocity. Further practical problems to be considered later should make it quite clear how the value of C can be found, given sufficient information, and we will first give a few more cases of the use of the rule.

There should be no difficulty with powers, such as

$$\frac{dy}{dx} = 5x^4 - 8x^3 + 5, y = x^5 - 2x^4 + 5x + C.$$

$$\frac{dy}{dx} = 7x^3 + 13x^2 - 9x, y = \frac{7}{4}x^4 + \frac{13}{3}x^3 - \frac{9}{2}x^2 + C$$

$$\frac{dy}{dx} = 8x^{1\cdot4}, y = \frac{8}{2\cdot4}x^{2\cdot4} + C = 3\cdot33.x^{2\cdot4} + C$$

$$\frac{dy}{dx} = 6x^{-3}, y = \frac{6}{-2}x^{-2} + C = -3x^{-2} + C$$

In dealing with roots and fractions, we must first express them by means of indices (or at least think of them as so expressed). Thus, if $\frac{dy}{dx} = 6\sqrt{x}$, we write it as $6x^{\frac{1}{2}}$ and then use the rule, obtaining $y = 6x^{1\frac{1}{2}} \div 1\frac{1}{2} + C$;

$$\frac{dy}{dx} = 6\sqrt{x} = 6x^{\frac{1}{2}},$$

$$y = \frac{6}{1\frac{1}{2}}x^{1\frac{1}{2}} + C = 4x^{1\frac{1}{2}} + C$$

$$(\text{or } 4x\sqrt{x} + C).$$

Similarly, if $\frac{dy}{dx} = \frac{8}{x^3}$, we think of it as $8x^{-3}$;

$$\frac{dy}{dx} = \frac{8}{x^3} = 8x^{-3}, y = \frac{8}{-2}x^{-2} + C = -4x^{-2} + C$$

$$(\text{or} -\frac{4}{x^2} + C).$$

As a first instance of the use of this process, and also as an instance of determining the constants that arise, we will consider the case of uniformly accelerated motion. Denoting by s, v, a, and t, distance, velocity, acceleration, and time respectively, and calling the initial velocity u, we have

$$\frac{dv}{dt} = a$$

so $v = at + C.$

But $v = u$ when $t = 0$, and substituting these values,

$$u = a \times 0 + C$$

whence $C = u.$

So $v = at + u,$

the well-known formula.

Again, $v = \frac{ds}{dt}$, so that

$$\frac{ds}{dt} = at + u$$

and consequently,

$$s = \tfrac{1}{2}at^2 + ut + D$$

where D is another constant. To determine D, we have that $s = 0$ when $t = 0$, and substituting these values,

$$0 = \tfrac{1}{2}a \times 0^2 + u \times 0 + D$$

whence $D = 0.$

Finally, then,

$$s = \tfrac{1}{2}at^2 + ut.$$

EXAMPLE 2. The gradient of a curve is given by $3 - 2x$, and the curve is to pass through the point (2, 6). Find its equation.

Here, gradient $= \frac{dy}{dx} = 3 - 2x$

Therefore $y = 3x - x^2 + C.$

But, to pass through the point (2, 6), we must have $y = 6$ when $x = 2$. Substituting these values,

$$6 = 3 \times 2 - 2^2 + C = 6 - 4 + C,$$

whence $C = 4.$

Thus, the required equation is

$$y = 3x - x^2 + 4.$$

EXAMPLE 3. As a next example, we will consider the suspension bridge, with a view to finding the shape of the curve in which the chain hangs, and the relations between the weight of the roadway, the sag, the span, and the tension in the chain.

In Fig. 97, AB is the roadway, CED the chain, and AC, BD, the piers. Let w be the weight of the road per unit length, and consider a portion of the bridge from the midpoint E to any other point P. Then, EO and PN being the verticals, $ON = x$, $PN = y$, tensions in the chain at E and P, T_0 and T respectively, the latter being inclined at an angle ϕ to the horizontal, we have for the equilibrium of the portion,

resolving horizontally, $T_0 = T \cos \phi$,
resolving vertically, $wx = T \sin \phi$

(wx being the weight of the road from O to N, which must be supported by the vertical component of T). Dividing,

$$\frac{T \sin \phi}{T \cos \phi} = \frac{wx}{T_o}$$

or, cancelling the T and remembering that $\frac{\sin}{\cos} = \tan$, $\tan \phi = \frac{w}{T_o} x$.

But $\tan \phi$ is the gradient of the curve at P, and so must be equal to $\frac{dy}{dx}$. Thus, we must have

$$\frac{dy}{dx} = \frac{w}{T_o} x$$

so that

$$y = \frac{w}{2T_o} x^2 + C.$$

But when $x = 0$, $y = y_0 = OE$, and substituting, we find that this makes $C = y_0$. The equation to the curve of the chain is thus,

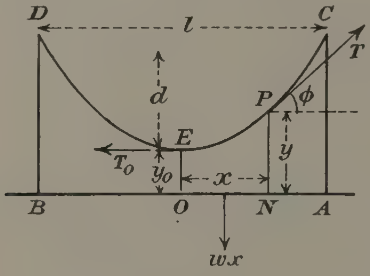


FIG. 97

$$y = \frac{w}{2T_o} x^2 + y_0$$

so that the curve is a parabola. To obtain the relations between sag and tension, etc., we go to the end C of the chain. If l is the span, $x = \frac{1}{2}l$ at C; also, if d is the sag, $y = d + y_0$ at C. Substituting these values,

$$d + y_0 = \frac{w}{2T_o} (\tfrac{1}{2}l)^2 + y_0$$

so,

$$d = \frac{wl^2}{8T_o}.$$

Knowing the sag and the span, as well as w, we can find T_o, and then the tension at any point.

EXERCISE No. 38

1. Write down the values of y for the following values of $\frac{dy}{dx}$: (a) $4x^3$, (b) $3x + 2$, (c) $x^4 - 3x^3 + x$, (d) $2x^{1.5} - 4x^{-0.6}$, (e) $\frac{1}{x^3}$, (f) $5\sqrt{x} + \frac{2}{\sqrt{x}}$.

2. If $dy = \frac{1}{2}x^3 - 2x$, and $y = 2$ when $x = 2$, find the formula for y, and the values of y when $x = 0$ and when $x = 4$.

3. The acceleration of a tain when starting is given by $4{\cdot}4\left(1 - \frac{t}{30}\right)$ ft./sec.², at t sec. after starting. Find formulae for the velocity and distance gone, in t sec., and calculate the speed and distance gone after 15 sec.

4. The sag of a suspension bridge is 16 ft., and the span is 80 ft. At the centre the height of the chain above the road is 4 ft. Find the equation to the curve, the tensions at the centre and the end, and the inclination of the chain at the end, given that the road weighs 2 tons/ft. (Remember that there will usually be *two* chains.)

5. In emptying a tank through an orifice at the bottom, it can be shown that $\frac{dt}{dh} = -\frac{k}{\sqrt{h}}$, k being a constant depending upon areas of tank and orifice, etc., t being time, and h the depth of the water. Taking $k = 60$, and the initial depth as 9 ft., find a formula for t, and deduce the time taken to empty the tank.

ANSWERS TO EXERCISE No. 36

1. 167, 130, 42, − 42, − 84, − 88, −84, ft./sec.².
2. $y = x + 4$; 45°.
3. 500 ft./min.; 11·36 h.p.
4. Radius = 2·233 in. = depth.
5. 7·746 ft. deep, 15·49 ft. wide.
6. $x = 5{\cdot}773$; 770 lb.-ft., nearly.
7. 0·5282, 0·5458.
8. $i = 0{\cdot}08944$ radians = 5°8′. $R_{min} = 35{\cdot}8$ lb.
9. Bottom and sides each 13 in. wide.

OUTLINES OF LOCOMOTIVE ENGINEERING

By A. Morton Bell, O.B.E., M.I.Mech.E., M.I.Loco.E.

LESSON IV

AUXILIARY APPARATUS

Regarding some of the more important subsidiary appliances required on a locomotive, pre-eminently, there are means needed for securing a supply of feed water to the boiler—both injectors and pumps are used. In the earlier locomotives, pumps driven from a moving part of the engine were the sole means employed. The injector replaced these pumps some years ago, owing to its simplicity and easy manipulation, and the fact that it would feed the boiler whether the engine was standing or running. Latterly, there has been a return to the pump, owing to the extensive use of feed-water heating. The early pumps having been mentioned, there is no necessity for further description here, but more details of recent developments will be given later.

Of injectors, there are two types—those operated with " live " steam, and those relying on " exhaust " steam. Both depend on a jet of steam moving at high velocity being suddenly condensed by contact with water moving at slow velocity, the momentum of the steam jet being, more or less, transferred to the water, thus producing a combined jet of sufficient resultant velocity to overcome the boiler pressure. The adjoining diagram, Fig. 12, will assist in elucidating this action.

The mixture of condensed steam and water enters the combining tube and then the diffuser or reversed nozzle. The cross-sectional area of the diffuser gradually increases and, in consequence, the velocity of the water flowing through the diffuser is diminished. By this means the pressure is gradually raised until it becomes sufficient to overcome the boiler pressure. The water is finally discharged to the boiler through a check valve, which prevents water flowing back through the injector when it is not in operation.

Refinements consist in methods of securing self-action in restarting should vibration of the locomotive in running cause operation to cease. This automatic restarting is secured by using nozzles with grooves turned circumferentially on their internal surfaces, or by hinged flaps which become deranged by the injector stopping and automatically fall again into correct position, when the requisite condensation is resumed.

Exhaust steam injectors are actuated by a supply of low pressure steam taken from the exhaust ; this is condensed on meeting the cool feed water, which it raises to a high temperature, causing the deposit of a considerable proportion of any solid matter it may contain. A jet of live steam is next introduced, which intensifies the velocity of the stream of hot water, driving it forward and ensuring its entry into the boiler. These injectors fulfil all the services of feed-water heaters, but there are those who prefer a separate, special, and independent method of heating the water without contact with the exhaust steam prior to its delivery to the boiler by a steam pump, the exhaust of such pump being utilized in the heater. An example of this arrangement is given in the accompanying drawing (Fig. 13), from which the different details can be readily followed.

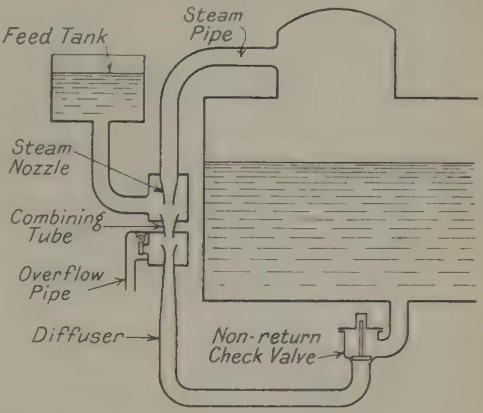
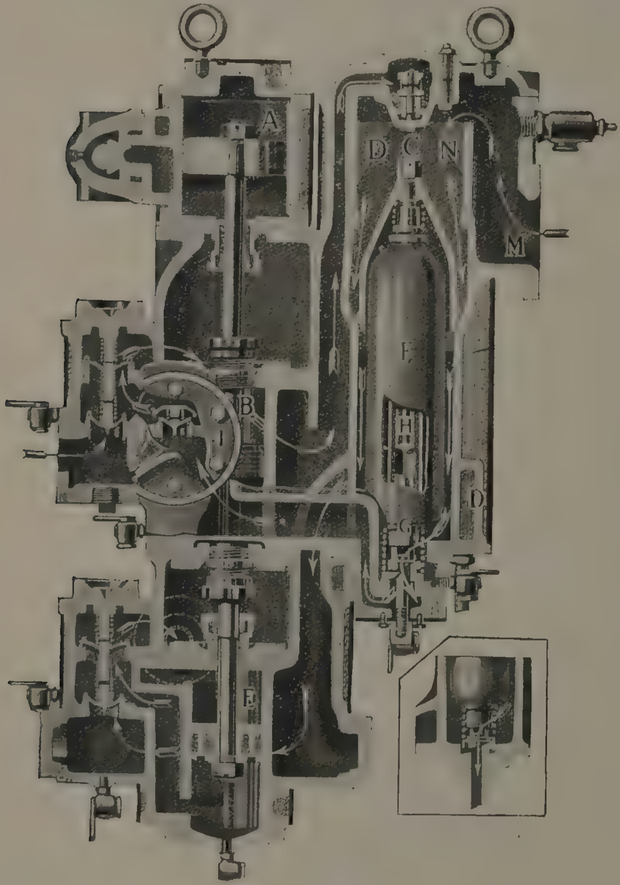


Fig. 12

There are two sources of waste heat on a locomotive—the escaping gases, and the exhaust steam. In attempts to utilize the former, mechanical difficulties are encountered which render any advantage still a doubtful gain. With exhaust steam, however, much less difficulty exists, and it can be utilized for heating the feed water, either on the suction side of the pump, or injector, used to feed the boiler, or on the delivery side, when the feed water can be heated in a tubular, or surface, contact apparatus, on its way to the boiler.

If the feed water supplied to a boiler working at 200 lb. per sq. in. pressure, normally at 60 °F., is raised to a temperature of 120° F. before delivery, a saving of 4 per cent in fuel may be anticipated, whilst if it can be heated up to 200° F., an economy of 12 per cent is possible. In addition to these savings, there are the subsidiary advantages accruing from the use of hot feed to be credited.



The feed water is heated in the preheater by mixing with the exhaust steam of the engine, and it is forced into the boiler by the pump. This steam passes through a stop valve, meets the exhaust steam of the feed pump, passes through an oil separator and enters the preheater *D* through opening *M*. The upper part of the preheater is then filled with exhaust steam at a pressure which is little less than that of the exhaust in the cylinders. The cold water is supplied from the tender by the cold water pump *B*, led to the spray valve *C* at the upper part of the preheater, where it mixes with the exhaust steam coming from the engine. This cold water condenses exhaust steam which is necessary for heating it, mixes with the water from condensation and falls to the bottom of the preheater, from where it is taken by the hot water pump *E*, and returned to boiler.

The oxygen contained in the water is released by preheating and goes into the air through the air exhaust vent *N*.

The relation between the cold water and hot water cylinders is such that the water-level in the preheater rises in consequence of the supply of additional water resulting from the condensation of the exhaust steam. At that moment the float *F* rises and uncovers the holes *G* of the guide rod *H*. The additional water passes through the holes *G* and the by-pass valve *I*. It is drawn by the cold water pump *B*, and returned to the preheater *D*. The pump is controlled by a throttle valve placed in the cab.

FIG. 13. SECTIONAL VIEW OF WORTHINGTON LOCOMOTIVE FEED WATER HEATER AND BOILER FEED PUMP

A very recent development for which some interesting claims are made is one wherein the feed water, heated by exhaust steam, is fed into the boiler by a mechanically operated pump, driven from a connection of the valve gear which is influenced by the degree of expansion, or cut-off, the locomotive is working in; that is to say, as the degree of expansion is increased the stroke of the pump is decreased, and the delivery of feed water to the boiler reduced automatically.

When it is considered that the actual return of useful work from the units of heat generated in the fire-box of a modern locomotive is only about from 4 to 5 per cent, it is not surprising to find many other ways of attempting to secure more satisfactory utilization of power have been suggested and tried. Methods of inducing better circulation of the water in the boiler to prevent accumulation of dirt, etc., on the heating surfaces are much in favour in the U.S.A. The sketch (Fig. 14) shows an adaptation of a device known as the "Thermo-syphon," consisting of tubes, or cavities, fitted to the fire-box, to quickly lead away the water receiving the most energetic heating. In this country more attention has been directed to the collection of dirt or deposit from the feed water prior to its entry into the boiler, thus ensuring cleaner tube and fire-box surfaces for more rapid conduction of heat. The drawing (Fig. 15) shows a very compact and efficient arrangement applied to a locomotive boiler. It consists of a drum, or dome, in which the feed is delivered from the injectors or pumps. Inside are a series of trays, or shallow pans, arranged in the form of a large spiral around a central pipe through which the boiler steam circulates. The feed water falling on this series of trays is directed along a tortuous path of spiral form, becoming highly heated on its descent to the boiler, and depositing the bulk of any solid matter it may contain on the trays and in the receiving chamber provided at the lower level; from this latter the mud and dirt can be easily withdrawn at intervals. In a similar manner, if the cover of the dome is removed, the shallow trays forming the spiral can be lifted out for cleaning. This arrangement is said to be particularly satisfactory in districts where indifferent water supplies are available, and the principle has been largely adopted in various forms. It may here be noted that at watering depots where a large number of locomotives are supplied, stationary plants for treating unsatisfactory water chemically, and "softening" it, are now frequently installed, and, there is no doubt, with good results. Good, clean feed water means much economy in fuel. The conductivity of the tubes of a boiler suffer to the extent of 25 per cent when they become coated with a scale of $\frac{1}{16}$ in. in thickness, whilst a covering of an $\frac{1}{8}$ in. thick means a loss of nearly 50 per cent. Further, if deposit and dirt can be kept out of a boiler and collected into some easily cleaned receptacle (as in the device shown in Fig. 15), there is every reason for the useful mileage of locomotives being much increased, due to the diminution in periods of stoppages for boiler cleaning, etc.

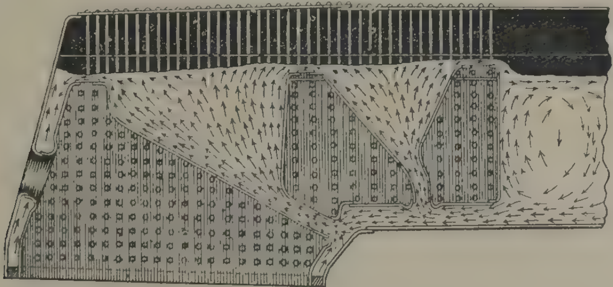


Fig. 14. Thermo-syphon in Fire-box

Conserving excess steam generated, or bulk of water heated, during intervals of standing, has also received attention, and engines have been fitted with receivers and reservoirs, wherein storage could be effected until such time as a maximum output was required. Such arrangements, however, all require plant and appliances which are apt to cause more trouble and expense in maintenance than they render good return for in practice, designers are therefore disinclined to adopt them unless some special or local condition favourably affects the problem, such as locomotives on mountain lines, wherein all power is required on the ascent, and the weight of the locomotive is needed only for assisting in deceleration and control on the descent.

All large boilers in power plants are fired mechanically nowadays, and, naturally, the growth in size of modern locomotives has drawn attention to the adoption of similar arrangements for firing on them; it is obvious that the amount of coal which can be fired by one man per hour is limited. The rate of combustion on a large goods engine equals some 100 lb. per sq. ft. of grate per hour, or, approximately, 1 ton of coal is required to be thrown on to the fire every hour.

The immense locomotives in the U.S.A. are now provided with such appliances, and many experimental installations have been made on British locomotives built for the Colonies and abroad. Briefly, there are two forms of stoker in service: one which relies on jets of high pressure steam to distribute the coal delivered on to the firing plate by the conveyor, and the other wherein a mechanical device imitates the action of the fireman's shovel and scatters the coal over the fire-grate: both are in extensive operation.

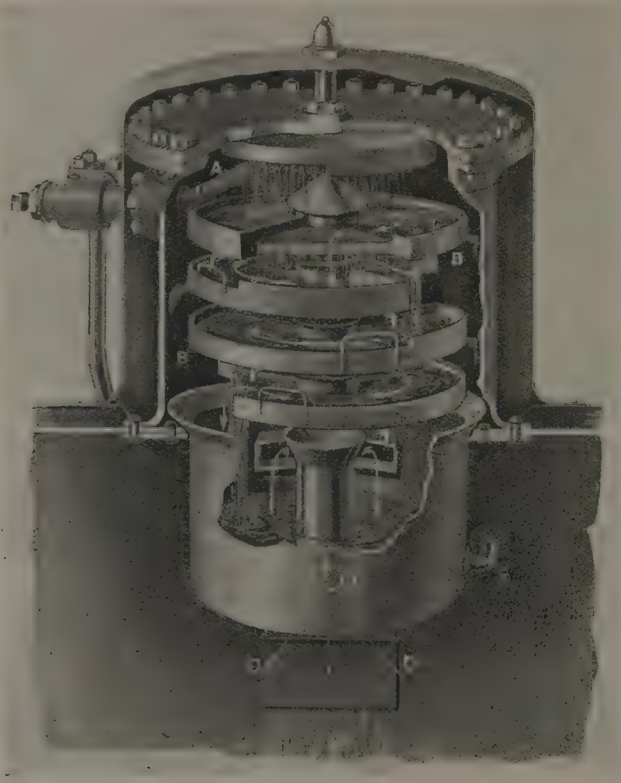


FIG. 15. "TITAN" FEED WATER HEATER AND PURIFIER

A. Feed inlet. *C.* Outlet for purified water.
B. Removable trays. *D.* Central steam passage.
E. Blow-off cock.

(Monarch Controller Co., London)

The coal for mechanically-fired locomotives is either crushed to uniform size at the depot before loading on to the tender, or passes a breaking-up device on its way from the bunker to the conveyor, which delivers the coal to the firing plate. The conveyor is operated by a small steam-engine controlled from the footplate, so that the delivery of coal can be regulated exactly to meet the requirements of the engine. As the firing is constant, other conditions can be adjusted to secure very efficient combustion. The diagram, Fig. 16, shows a complete installation on engine and tender.

Another method of utilizing coal of inferior grade, which has found limited adoption chiefly in countries where the fuel available has, more or less, favoured its use, is that known as "pulverized" coal. The coal is ground to a very fine powder, dried and carried in a perfectly tight bunker. It is conveyed by a helical conveyor driven by a steam motor, through flexible piping to the furnace, where it is injected and distributed by low pressure air blast furnished by a blower. The fire-box is specially arranged, and the blowing apparatus is worked in conjunction with the conveyor, used to transport the powdered coal from the tender to the fire-box. The fuel is burned without residue, so that the useful period of a locomotive in service can be extended by the elimination of periods for cleaning the fire, etc. Objection to the system exists, however, in the risk of explosion, spontaneous combustion, etc., in the storage of the powdered fuel.

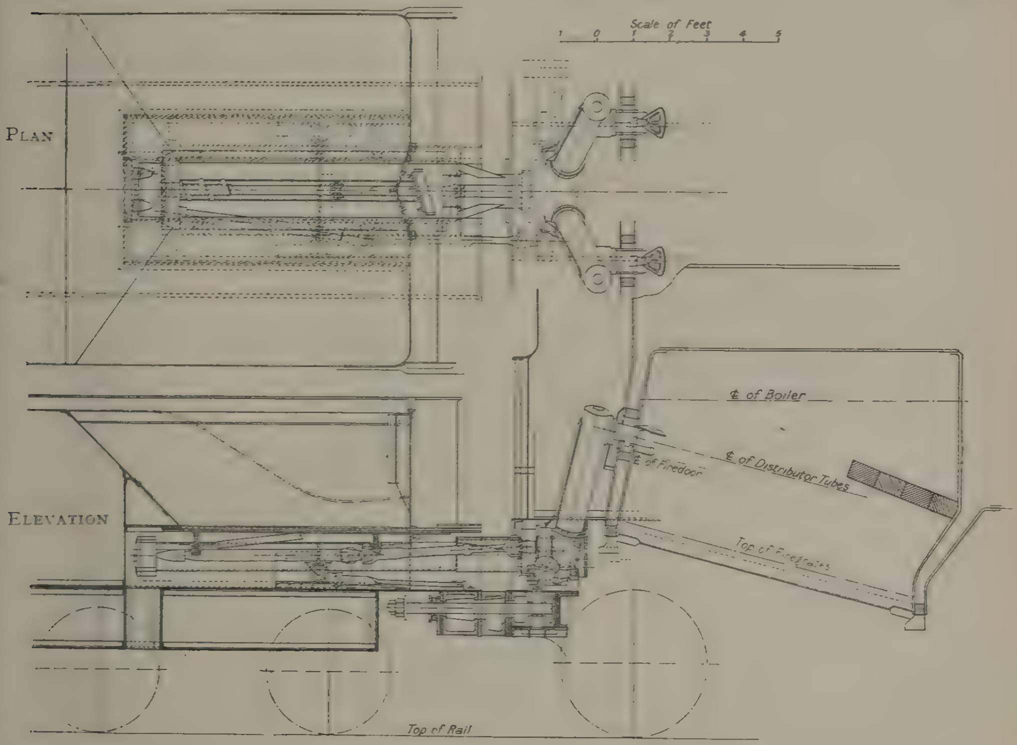


Fig. 16. Arrangement of Mechanical Stoker

Peat can be similarly powdered and burnt.

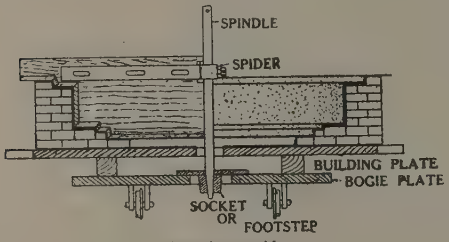
PATTERN-MAKING AND FOUNDRY WORK

By J. McLachlan and C. A. Otto

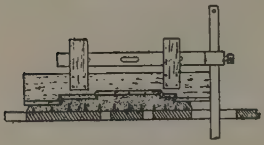
LESSON XI

LOAM MOULDING

Loam moulding is more complicated and difficult than it is sometimes thought to be by those whose experience has been confined to dry and green sand moulding. Loam may be used to build a complete mould, including cope and cores, but sometimes it is better practice to construct the mould partly of loam and the remainder of sand. Practical reasons, such as the difficulty of locating small loose parts of a pattern in a swept-up loam cope, render it advisable at times to make a loam pattern which can be used for making a mould either in loam, dry or green sand.



Sweeping mould



Sweeping cover

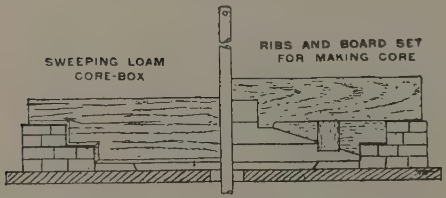


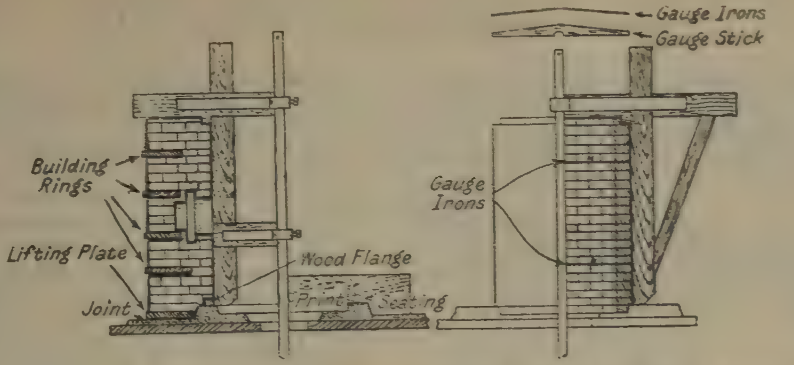
Fig. 47. Sweeping Shallow Work About a Vertical Spindle

The value of loam is most evident in making moulds for cylindrical castings, because this work offers facilities for sweeping about a vertical or horizontal spindle, and boards or sweeps form the principal medium for obtaining the required shape. It is in the making of moulds for cylindrical castings that loam possesses economic advantages. A cheap profiled edged board takes the place of a costly pattern. It must be remembered that when many similar castings are required, it is generally better to supply a full pattern because sand moulds can be more quickly produced, and the saving of time on each mould would compensate for the extra cost of pattern-making if a sufficient number of castings are wanted.

Materials Used in Loam Moulds. A loam mould involves the use of a base-plate, upon which the mould is built; a spindle, when the work is cylindrical; grids to strengthen the mould and allow the parts to be lifted; and bricks to support the loam. Three loam mixtures are generally used and are known as building, facing, and core loam mixtures. The building loam is used for interposing between the bricks, the facing loam, as its name implies, forming the face of the mould, while the core loam is more open in texture. These mixtures compare with facing, backing, and core sands used in green or dry sand moulding. Owing to the strength of loam mixtures, it is necessary to use openers, so that the loam may be sufficiently porous at the time of casting. For this purpose, chopped straw, horse dung, or horsehair may be used. Many foundries reduce the strength with ground coke or sharp sand. Strong facing mixtures crack considerably on drying, and great care has to be exercised later in filling these cracks to avoid scabbing the casting.

The majority of loam moulds are for cylindrical castings, and are prepared either about a vertical or horizontal spindle, according to the kind of work required; it will be as well, therefore, if the two forms are considered separately.

Sweeping About a Vertical Spindle. This form of sweeping is adopted when the diameter is large compared with the length of work. The spindle and socket are the most important parts of the equipment. The spindle should be accurate to size and perfectly straight. The most convenient sizes are 1½ in. and 2 in. for



SWEEPING COPE

SWEEPING PATTERN FOR USE IN MAKING SAND OR LOAM MOULD

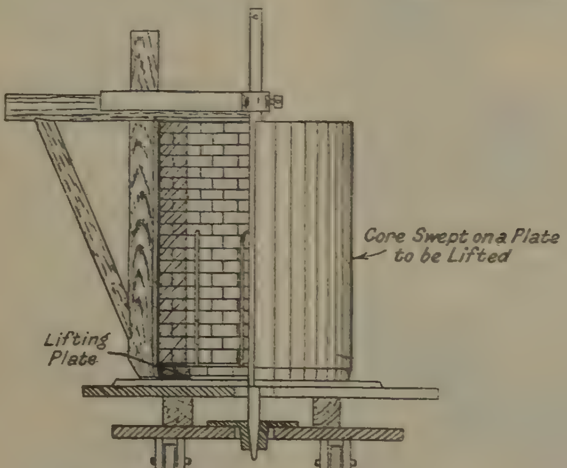


FIG. 48. SWEEPING DEEP CYLINDRICAL WORK

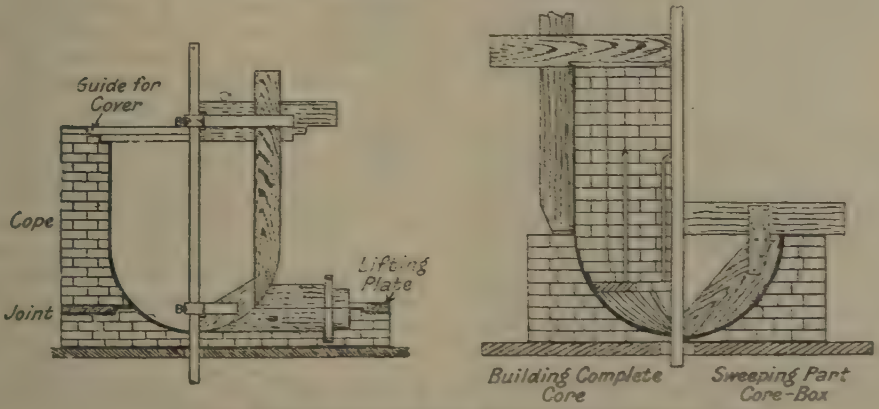


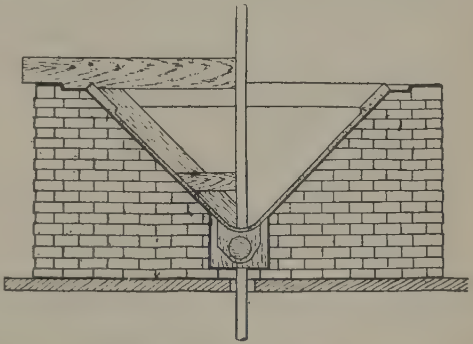
FIG. 49. SWEEPING CYLINDRICAL WORK HAVING SPHERICAL END

small moulds, and 3 in. diameter for medium and large moulds, excepting when the sweep and other equipment are very heavy, when a 4 in. spindle will give greater rigidity. A spider or tail-piece is made to fit on the spindle to which the sweep can be attached. Much difference of opinion exists respecting this form of equipment. Some prefer wall braces at the top of the spindle, while others depend on a long tapered fit in the socket. The spindle may be fixed in the socket and the sweep rotated about it, or the sweep may be fixed to the spindle and rotated with it. The latter is in more common use, and is the method illustrated in this article. With this form of equipment, and a suitable base or building plate, moulds may be swept direct, or a loam pattern may be prepared to be used in the same manner as a wood pattern. When the mould is swept, the operation is frequently referred to as sweeping the cope. Subsidiary pattern sections are made in wood, that can be located in their relative positions during the time of sweeping the mould, or, in the case of a loam pattern, after it has been dried. When there are many pattern sections in wood it is the more general practice to sweep a loam pattern, because the sections can be located more accurately on it. Generally, such a pattern can only be used once, another pattern being swept for the second mould.

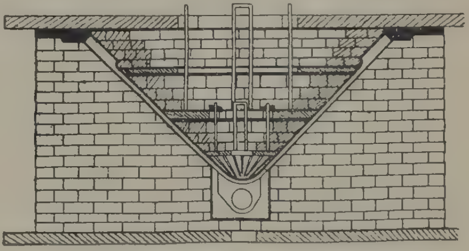
Cylindrical cores that are large in diameter are prepared about a vertical spindle in exactly the same way as a loam pattern, excepting that more care is necessary because it is required to resist the pressure of the metal and be free venting. When the core is practically straight the whole can be swept; it may, however, be impossible to sweep it wholly in this way. In such cases a seating or part core-box may be swept and the core built in, and any remaining shape completed by means of a separate sweep. Instead of adopting either of these methods, a thickness may be swept on the mould, corresponding to the thickness of metal in the required casting. This forms a complete core-box in which the required core can be made. This latter is of considerable value in certain kinds of work, as a regular thickness is assured and the actual working space is much less than when making the core separately.

Horizontal Sweeping. When the work is small in diameter compared to its length, and loam can be conveniently employed in the preparation of the mould, it is swept horizontally. The equipment necessary for this method differs from that used in vertical sweeping. It is true that a spindle is used, but this may be solid and parallel, or it may support a barrel according to the kind of work and the form of sweeping.

Racks are required to supply a bearance at each end of the spindle. These vary in size from small bench racks to very large racks that are mounted on wheels, and are capable of adjustment to suit varying lengths. As in vertical spindle work, the horizontal spindle may be used for making the mould direct, for making a loam pattern or a core. The method is more commonly associated with the making of cores, as, even in foundries where dry sand is preferred for the general run of work, loam is used for many of the plain cylindrical cores.



Mould swept and sweeping a thickness



Building core

Fig. 50. Using Thickness of Loam on Mould Corresponding to Thickness of Metal

When the core is comparatively small and can be handled, a solid spindle is generally used, unless it is fairly long and the spindle is required to give support during the time of casting. Many cores, however, need to have the spindle removed after they have been swept and dried. In such instances, some form of rodding is necessary to give the requisite strength to the core. If the spindle can be left in the core it is generally hollow and perforated with holes, so that the ends of the spindle, or barrel, as it is called, will permit the ready escape of gases generated at the time of casting. For sweeping cores or loam patterns, the wood sweep is fixed to the racks at the required distance from the centre of the spindle, and the spindle is rotated in its bearings. When the spindle is to be removed from the core, a layer of straw rope is first wound on the spindle, and the shape gradually made up from straw and loam until within about ¾ in. of the working edge of the sweep when loam only is used to complete the shape. In other cases, when the spindle or barrel remains in the core, loam and straw rope are applied to the barrel together, and the shape of the core gradually formed as the barrel is rotated. The straw rope serves a dual purpose, as it supports the loam and facilitates the escape of gases from the core. When the spindle is to be withdrawn the straw rope carbonizes during the time of drying the core, leaving the spindle loose. With the perforated barrel the straw makes a contact with the vent holes.

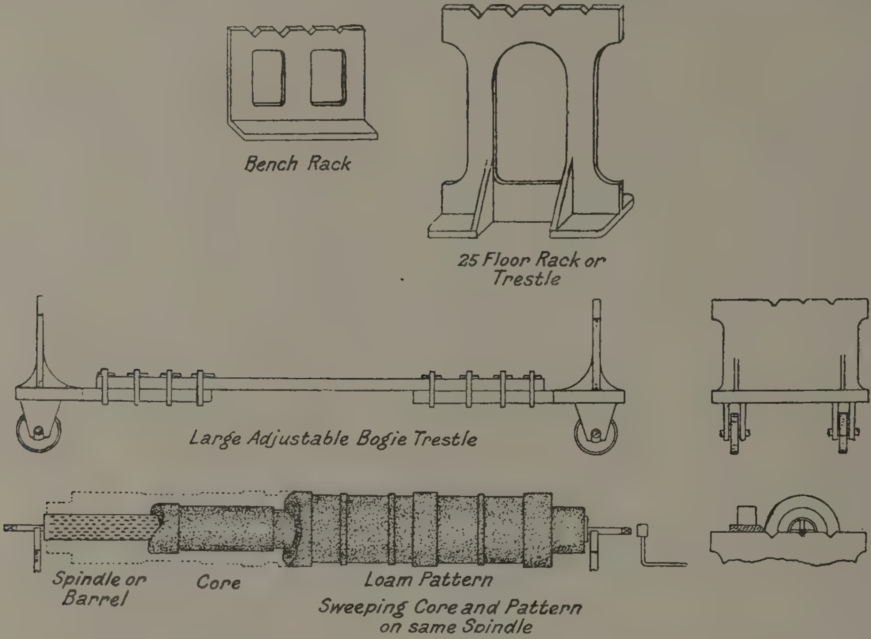

Bench Rack
25 Floor Rack or Trestle
Large Adjustable Bogie Trestle
Spindle or Barrel
Core
Loam Pattern
Sweeping Core and Pattern on same Spindle

FIG. 51. SWEEPING LOAM WORK ON A HORIZONTAL SPINDLE OR BARREL

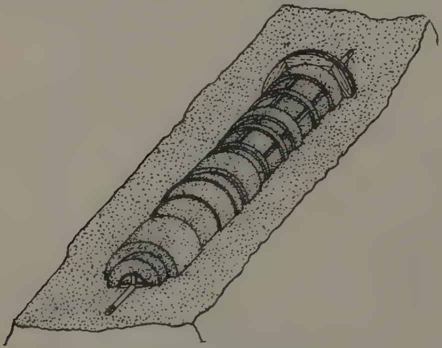
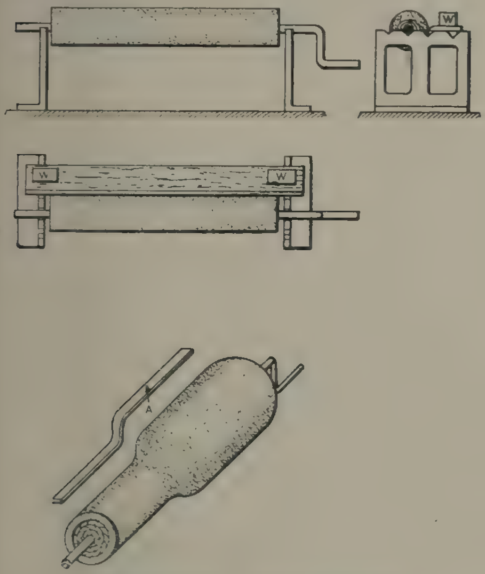


FIG. 52. LOAM PATTERN BEDDED INTO SAND FLOOR

Wood wool or hay may be used instead of straw rope. A loam pattern may be made in exactly the same way, but it is only necessary that it should have a reasonably smooth surface, and be capable of withstanding the ramming of sand about it when used in making the mould. Sometimes, when only one barrel is available for pattern and core, the core is first swept, then dried, after which a thickness of loam is applied to it to form a pattern. When the mould has been prepared the thickness is removed, and the core coated with a face wash to be dried preparatory to setting in the mould. Subsidiary pattern sections can be attached to a loam pattern swept horizontally, and such a pattern can be used many times as long as it is not broken when stripping.



Supporting irons like *A* are bedded into loam

FIG. 53. SWEEPING SMALL CYLINDRICAL CORES WHEN SPINDLE IS TO BE REMOVED

Sweeping moulds horizontally is a method not frequently adopted, because full cylindrical moulds would need to be made in halves and this involves elaborate equipment to ensure accuracy. The sweep is secured to the spindle by means of two or more spiders, and rotated with the spindle. Bricks form the main support for the loam in this instance, but a grid, frame or box is necessary to enclose the brickwork. A box is preferable if the work is to be cast on end to ensure a sounder casting, because of pressure of the metal when casting.

While it is true that the majority of loam jobs are prepared about a spindle, there are a great many other jobs made in loam without their use. In fact, the number of ways in which loam may be applied in the making of moulds is limited only by the moulder's ingenuity. Sometimes this ingenuity is exercised at the expense of economy. The profile of a required casting or its core can be formed frequently on a board, and used against a guide which defines the contour of a shape required, enabling a core to be formed or a mould made. A certain amount of pattern work is invariably necessary, but, as a rule, very much less than would be necessary for green or dry sand work.

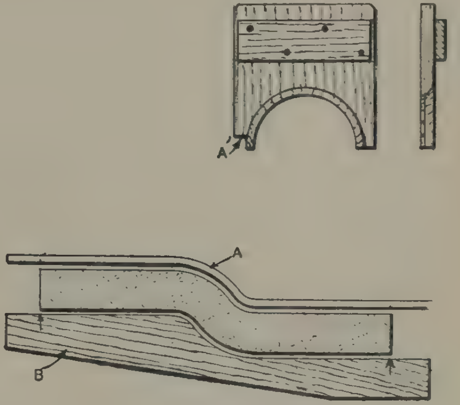


FIG. 54. HALF CORES MADE WITH STRICKLE CUT AT "A" TO FOLLOW WOOD OR IRON GUIDE

All loam work must be dried and special drying facilities are required in the form of large ovens, into which work can be moved on bogies. Surfaces against which the metal comes into contact need a special facing wash.

MACHINE TOOLS

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XIX

MILLING MACHINES—(*contd.*)

Vertical Spindle Machines. These are built after the general model of the drilling machines, with a vertical spindle, holding edge mills most frequently. Face cutters may also be used with these machines when desired. They are also adapted for profiling irregular outlines with edge mills under the control of formers. Linear and circular tooling, external and internal, are performed with the work carried on compound tables. Edges can be tooled, and faces without resetting the work. The slotting machine is superseded on convex and concave faces, because the slotting tool takes many separate cuts with lost strokes, and leaves minute facets, which objections do not apply to the milling cutter. The lateral stress in edge milling is severe, but this may be resisted with a steady bracket that supports the spindle close to the cutter.

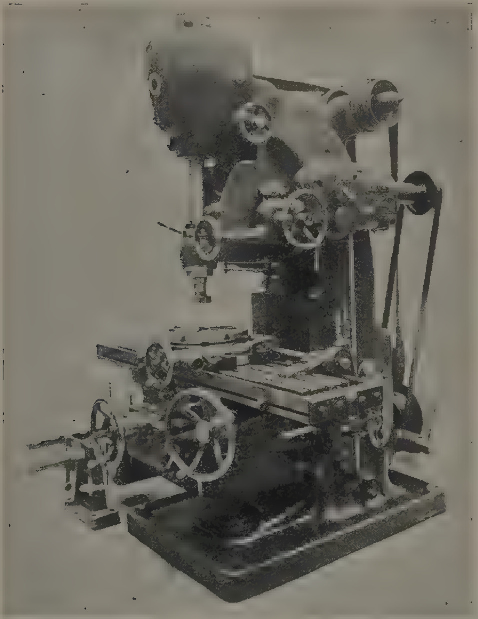


Fig. 132. Heavy-duty Vertical Miller

An example of a heavy duty machine is shown by Fig. 132, by Smith & Coventry, Ltd., of Manchester. The spindle, 3 in. diameter, is driven from a constant-speed pulley making 375 r.p.m., put in and out of connection with a friction clutch. Twelve spindle speeds are provided, ranging from 20 to 500 r.p.m.—the six faster being obtained direct from a belt passing over guide pulleys, the six slower through gearing, lever-operated. The lower end of the spindle is supported close to the cutter with a bearing sliding with it on the face of the column, with a vertical range of 7½ in., and balanced. The spindle has 12 changes of feed, from 1 to 20 in. per min. The table has a working surface of 58 in. by 16 in. It is carried on a knee on which it is adjustable vertically with a hand crank, bevels, and telescopic screw. It has 12 rates of self-acting feed, in longitudinal, cross, and vertical directions, ranging from 1 to 20 in. per min., independent of the spindle speeds, and changed with the movement of a lever on a gear-box. The design is such, that a quarter revolution of the lever in a clockwise direction gives one rate finer of the feed, while the opposite movement gives one rate coarser. The gears in the feed-box are of hardened steel, and run in oil, supplied as long as the machine is in operation, but passing afterwards to the main driving gears. A tank holds the cooling liquid, whence a jointed pipe supplies it to the cutter. It is returned to the tank through a strainer, and flexible tube.

The Plano-millers. The most striking advance in milling machine design during the present generation has been that of the plano- or slab-millers. These were introduced as a challenge to the reciprocating planers, but they have developed into a large group apart, having little in common with their prototypes beyond the barest outlines. That is, they have a long bed carrying a travelling work table, upright housings supporting a cross rail with a tool carriage or carriages, and with or without side tool heads.

The difference between the earlier and later machines is immense. For some tasks the plano-millers fear no rivalry from the planers. Gang mills will tool wide irregular faces simultaneously that would be very tediously done with single-edged tools. And two, or three separate sets of tools can be in operation at one time, on the same or on separate adjacent faces. Very heavy slogging is performed with deep cuts and coarse feeds, by the lavish employment of inserted cutters of high speed steel, and latterly of stellite, in heads of large capacities that cover a width of several inches.

Variations in design are numerous. The term plano-miller is misleading when applied to those which are vertical spindle machines only, though they do not resemble in build that class. In the typical machine the cutter arbor is rotated between bearings that have lateral adjustments along a cross rail, which is adjustable vertically on the faces of upright housings that flank the bed. Arbors of different sizes and lengths interchange. Geared changes are provided for rotational speeds. Variable rates of feed are imparted to the work-carrying table.

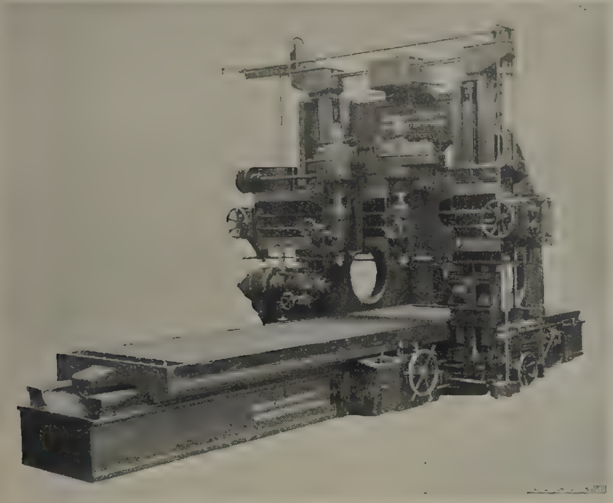


Fig. 133. Three-spindle Plano-miller

Fig. 133 illustrates a recent machine by Smith & Coventry, Ltd., with two vertical spindle heads for face mills, and one side head for face mills, or, using the tail support for an arbor, for edge cutters. The capacity is for work up to 8 ft. long, 5 ft. 6 in. wide, and 4 ft. 6 in. high. A feature to be noticed is that the controlling levers and arrangements are duplicated, so that the attendant can operate the table movements, the cross slide, cutter heads, and spindles, from either side, which avoids loss of time in walking round a big machine.

The bed is deep, of a box section, with the top surface enclosed, and having flat ways with square gibs to receive the work table. The latter with a working surface of 8 ft. by 5 ft. has tee slots for holding work, and waste lubricant pockets at each end. When cross-cutting is being done, the table is locked. The housings, of box section, and semi-parabolic outline are bolted to the sides of the bed. They carry the cross slide with square gibs, which receives the two cutter heads, each fitting with a narrow guide, and taper adjusting strips, and provision for locking. The heads have movements independent of each other, and their spindles have a range of 12 in. The side head has vertical and horizontal adjustments, and the support for the tail of the arbor can be aligned with it.

The machine is driven with a 20 h.p. motor, on the farther side of the bed, driving through a centrifugal clutch and a gear-box that gives nine rates of automatic traverse to the table in either direction, at speeds ranging from 1 in. to 8 in. per min. The table is driven with the Sellers' worm and rack, motion being transmitted to it from the gear-box through hardened steel spiral bevel gears running in an oil bath. The table traverse gears are controlled by the levers seen at the foot of the right-hand upright. Rapid traverses of the table at a constant speed of 4 ft. per min. are controlled with a lever in the box at the side of the bed; hand traverse, for setting work, being effected with the large hand-wheel adjacent. When either is in action, the other is thrown out automatically.

The vertical spindles are driven through a six-speed gear-box by a series of vertical and horizontal splined shafts, and hardened steel spiral bevel gears to a worm reduction gear inside each head. These give a range of fast spindle speeds. Slow speeds are obtained through a back gear located in each head between the worm reduction gear and the spindle. The small hand-wheel at the side of the head operates the back gear, bringing it into action through an eccentric sleeve. Rapid power traverse is applied to the cutter heads and cross-slide, operated with levers and friction clutch connections. Hand traverses are effected with the large hand wheels in front of the cross-slide. Dial indicators and pointers are lifted to the traverse screws of both cutter heads, and graduated scales and pointers enable the movement of both of the table and cutter heads to be calibrated.

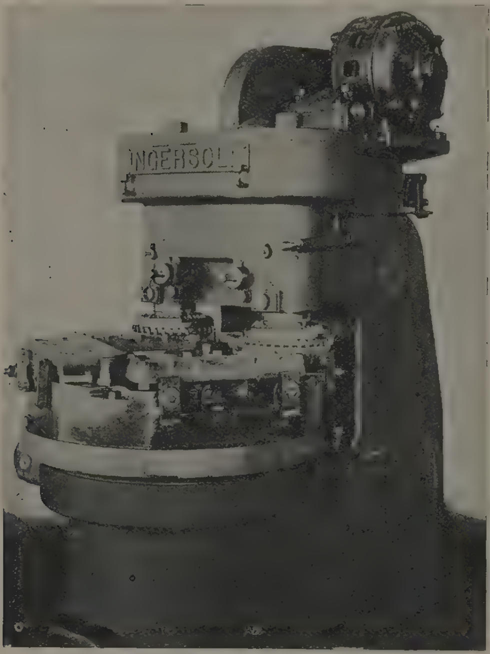


Fig. 134. Cylinder-head Castings being Surfaced on a Two-spindle Rotary

The cutter spindles are 4 in. in diameter on the driving part, and have an 8 in. flange at the bottom end for the attachment of large face milling cutters. The spindle nose has a No. 14 B. and S. taper, and a central hole receives a draw bolt. A 9 in. sleeve carries the spindle, adjusted vertically through a hand-wheel worm wheel, rack and pinion gears. The spindle runs in a conical bearing at the bottom and a parallel bearing above.

Although the first machines in this great group were primarily designed for the employment of edge mills, either singly, or disposed in gangs on an arbor, the face mills are used now far more extensively when plane surfaces are being dealt with. They are not so liable to spring and they cut more deeply. Invariably they have inserted teeth, set at an angle of front rake for incisive action, and spaced to give ample chip clearance. The cutters are well supported on their heads, and they often cover a very large area. With vertical spindles, either face or edge mills can be used. And as with the simple vertical-spindle machines, circular milling is done, the work being carried on a circular table, like that shown in Fig. 132, mounted on the main table that reciprocates. And profile milling is also accomplished on these machines, as on the vertical spindle type. Fig. 134 shows two face cutters working on cylinder head castings in fixtures on a rotary table machine.

Plane horizontal, vertical, and angular faces are more often tooled with face cutters than with edge and gang mills, two or three face tools being used for surfaces that are adjacent. The edge mills are chiefly reserved for producing outlines that are curved in section. When these are wide it is usual, if the surfaces are not continuous, to fit a central support on the cross rail between the arbor bearings. Since the work of tooling broad surfaces sets up severe stresses on these machines, their framings and entire fittings have to be made substantial enough to absorb all vibrations, which would cause chatter, and vibration, and produce inaccurate work. The cost of cutters is high, but they can be reset and reground through a long life.

Rotary Planers. These are face milling machines, with horizontal spindles carrying large heads with inserted cutters, used for facing work, fed past on a travelling table, or, the head is fed past the work on a fixed table. The design is old. It is valuable chiefly in constructional work, for planing the ends of rolled sections, and also in engine building, for tooling the feet of standards and beds, etc. There are perhaps a dozen distinct variations on this simple design.

ELECTRO-TECHNICS FOR MECHANICAL STUDENTS

By A. P. Young, O.B.E., M.I.E.E.

LESSON XI

ALTERNATING CURRENT APPARATUS AND SYSTEMS

Alternating current (A.C.) is widely adopted for the distribution of power and light, mainly because of the ease with which electrical energy transmitted with this system can be transformed from one voltage to another. Electrical energy can be transmitted over long distances by the A.C. system of distribution with a considerable saving in cable cost and transmission losses, because of the high voltage that is permissible.

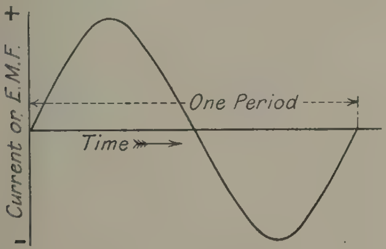


Fig. 76. An Alternating Current Wave.

Theory of Alternating Currents

(*a*) **Alternating Current and E.M.F.** An electric current which passes through a complete cycle of changes in magnitude and direction at regular intervals is called an alternating current. When dealing with the principle of operation of the D.C. generator, it was explained that the induced E.M.F. in the armature windings was alternating in character, as illustrated in Fig. 51, on page 938. The current generated by such an E.M.F. would also alternate, first flowing in one direction and concurrently changing in magnitude, and then reversing its direction with similar changes in magnitude. A complete cycle is shown in Fig. 76, and constitutes a *positive* half cycle and a *negative* half cycle. The time taken for one complete cycle of changes is called a *period*, and the number of periods per second is the *frequency*, or, as it is sometimes termed, the *periodicity*, represented by (~). The frequencies adopted for A.C. supplies in this and other countries vary from 25 to 100 cycles. The most common frequency, however, is 50 cycles.

(*b*) **Equivalent A.C. and D.C. Values of Current.** Since an alternating current is constantly varying in amplitude, it is necessary to decide upon some numerical value of current, which shall be equivalent in effect to a continuous current of the same value. The heating effect of a current at any instant depends upon the *square* of the current value at that instant. Therefore, if an alternating current is to produce the same heating effect as a continuous current, the mean value of the square of the current over a period must be equal to the square of the continuous current. The numerical value chosen is, therefore, the *root-mean-square*, or, abbreviated, the R.M.S. value. This applies to both E.M.F. and current.

(*c*) **Sine Law Wave Form.** In dealing with A.C. problems, it is assumed that the wave forms of E.M.F. and current follow the simple sine law and, in general, this assumption is sufficiently accurate. It is also found in practice that the wave forms of alternating E.M.F.'s and currents closely approximate the simple sine wave. On the basis of this assumption, the ratio

$$\frac{\text{R.M.S. value}}{\text{maximum value}} = \frac{1}{\sqrt{2}} = 0{\cdot}707.$$

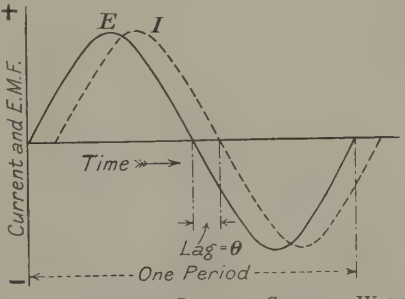


Fig. 77. Showing Lagging Current Wave for an Inductive Current

This ratio is termed the *amplitude factor*, and, knowing the R.M.S. value, we can easily determine the maximum value reached by the alternating wave.

(*d*) **The Effects of Resistance, Inductance, and Capacity in an A.C. Circuit.** In order to determine the current flowing in a D.C. circuit it is only necessary to consider one factor, viz., resistance, when a given E.M.F. is applied. With A.C. circuits, however, the problem is more complex, owing to the introduction of two other factors, viz., inductance and capacity. Not only do these factors affect the magnitude of the current, but they also cause the current wave to be displaced in relation to the E.M.F. wave applied to the circuit.

The constantly varying current flowing through an inductance results in the generation of a counter E.M.F. of self-induction, which is displaced in time relative to the current wave by a quarter of a period (90°). If we regard one complete cycle as 360°, then *the self-induced E.M.F. wave* is displaced by 90° from the current wave, and the current changes are thus caused to occur later than those of the applied E.M.F. The current is then said to be *lagging* in phase, and Fig. 77 shows the phase relationship between the applied E.M.F. and current waves in a typical inductive circuit.

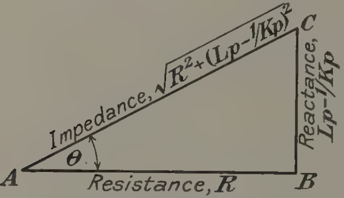


FIG. 78. GRAPHICAL METHOD OF DETERMINING PHASE DISPLACEMENT

Electro-static capacity, as, for example, in the form of an electric condenser, results in a leading current, i.e. the changes in current occur earlier in time, in respect of the changes in the applied E.M.F.

We will now consider the separate and combined effects of resistance, inductance, and capacity.

1. **Resistance.** If the circuit be one of resistance only, then the current and E.M.F. changes will be in phase, and the current in the circuit will be determined in the same manner as for a D.C. circuit, thus—

$$\text{Current } (I) = \frac{\text{applied E.M.F. } (E)}{\text{resistance } (R)}$$

2. **Inductance.** For a circuit in which the inductance is so large that, by comparison, the resistance is negligible, the current is computed from the equation

$$I = \frac{E}{Lp} \quad . \quad . \quad . \quad . \quad . \quad (35)$$

Where I = current in amperes.
E = applied E.M.F. in volts.
L = self-induction in henries.
p = $2\pi \times$ frequency (f).

The quantity Lp is usually called the *inductive-reactance.*

The effect of the inductance, assuming the resistance of the circuit to be negligible, would be to cause the current changes to *lag* behind those of the applied E.M.F. by a quarter of a period or 90°.

3. **Combined Resistance and Inductance.** To take the more general case of a circuit consisting of both resistance and inductance, the current is given by the equation—

$$I = \frac{E}{\sqrt{R^2 + L^2p^2}} \quad . \quad . \quad . \quad (36)$$

The quantity $\sqrt{R^2 + L^2p^2}$ is termed the *impedance* of the circuit, and the tangent of the angle of lag will be proportional to Lp/R. When the resistance is large in relation to the reactance the phase displacement of the current wave relative to the applied E.M.F. wave will be small, and vice versa.

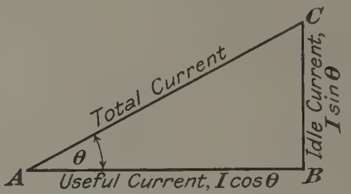


FIG. 79. GRAPHICAL REPRESENTATION OF USEFUL AND IDLE CURRENT IN AN A.C. CURRENT

4. **Capacity.** The current flowing in an A.C. circuit comprising capacity only is given by the equation

$$I = E.K.p = \frac{E}{\frac{1}{Kp}} \quad . \quad . \quad . \quad . \quad (37)$$

Where K = capacity in farads,
and p = $2\pi f$.
$1/Kp$ is called the *reactance* of *capacity*.

Capacity has the effect of causing the current to *lead* in phase relationship with the applied

E.M.F., and, in the extreme case of a circuit of capacity only, the current wave would be 90° in advance of the applied E.M.F. wave.

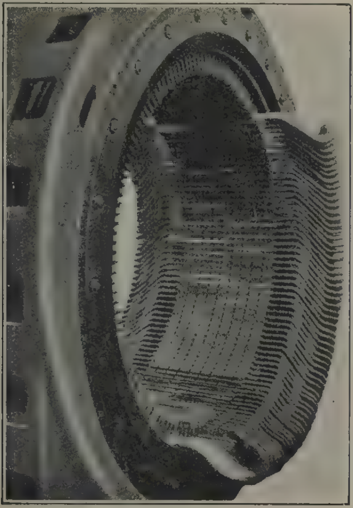


FIG. 80. PARTLY-WOUND STATOR FOR 1,500 K.V.A., 11,000 VOLT ALTERNATOR

5. **Combined Resistance, Inductance, and Capacity.** For a circuit consisting of all three factors in series, the current flow is determined from the equation—

$$I = \frac{E}{\sqrt{R^2 + (Lp - 1/Kp)^2}} \quad . \quad . \quad (38)$$

The reactance of the circuit would be that due to the inductance and capacity, and is represented by $(Lp - 1/Kp)$. The impedance will be the quantity $\sqrt{R^2 + (Lp - 1/Kp)^2}$. Since the effects of inductance and capacity on the phase displacement of the current are exactly opposite, it will be evident from equation (38) that if the reactances of inductances and capacity are equal, then the expression $(Lp - 1/Kp)$ becomes zero, and the circuit is equivalent to one of resistance only, in which case the current and applied E.M.F. will be in phase.

The angle of lead or lag will be such that

$$\text{Tan } \theta = \frac{Lp - 1/Kp}{R}$$

and if $Lp > 1/Kp$, the current will lag behind the applied E.M.F., whilst if $1/Kp > Lp$, then the current will be a leading one.

The effect on the phase displacement of varying the three factors may be studied in a graphical manner, by drawing a triangle to scale according to their respective values, as shown in Fig. 78. The base (AB) represents the resistance of the circuit and the vertical side (BC) the combined reactance, due to inductance and capacity. The hypotenuse (AC) then gives the impedance of the circuit, and the angle θ is the angle of phase displacement between the current and the applied E.M.F.

POWER IN AN A.C. CIRCUIT

In speaking of the current in an A.C. circuit, it is convenient to consider the total current as divided into two components, viz.—

1. The useful current which is in phase with the applied E.M.F. and expressed mathematically as $I \cos \theta$, and
2. The idle or wattless current, which is 90° out of phase with the applied E.M.F., and expressed as $I \sin \theta$.

Shown graphically as in Fig. 79, these two components are represented by the base (AB) and perpendicular (BC) respectively, whilst the total current is given by the hypotenuse (AC).

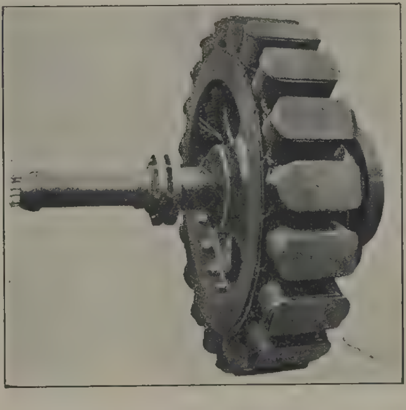


FIG. 81. COMPLETE ROTOR AND SHAFT, SHOWING FIELD WINDINGS AND SLIP RINGS

The power in an A.C. circuit will then be the product of the useful current and the applied E.M.F., and is expressed as

$$\text{Power} = E.I. \cos \theta \quad . \quad . \quad . \quad (39)$$

Cos θ is called the *power factor* of the circuit, and in order that any A.C. system shall operate efficiently, it is important that the power factor be as near unity as possible.

Alternators

The modern alternating current generator or alternator essentially consists of

1. A stationary armature or *stator*, and
2. A rotating field magnet system called the *rotor*.

It is now general practice to rotate the field magnets rather than the armature windings, because this simplifies the construction, and, furthermore, with the high voltages now universally adopted with A.C. systems, stationary armature windings can be more satisfactorily insulated. Another important point is that the current generated is taken direct from the stator terminals without the necessity of sliding contacts, such as slip rings.

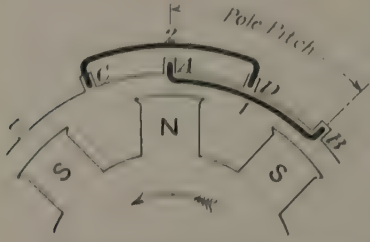


FIG. 82. SINGLE AND TWO-PHASE ALTERNATOR WINDINGS

The field magnets are energized by a low-voltage, direct-current generator, called the *exciter*, usually direct coupled to the alternator shaft. The current is supplied to the magnet coils through carbon brushes bearing on insulated metal rings on the alternator shaft, and connected to the magnet coils.

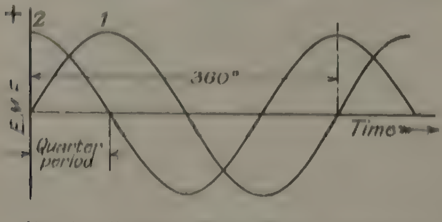


FIG. 83. SHOWING PHASE RELATIONSHIP OF TWO-PHASE ALTERNATING E.M.F.'S

The armature core consists of thin, soft iron laminations securely fixed to the stator frame, and slots are provided into which the armature coils are fitted. Fig. 80, showing a partly wound stator, will serve to illustrate present-day stator construction. The number of slots per pole is usually six, in order that the same design of lamination may be used for either single, two- or three-phase alternators. Further reference will be made to this later. In some cases, however, twelve slots per pole are provided.

The rotating magnet system for low-voltage, low-speed alternators, usually comprises a series of field magnets, mounted on a cast-iron yoke, as shown in Fig. 81. In this case, the pole cores are of solid steel, but for medium and high voltage machines laminated poles are employed.

The armature windings of a direct-current generator are uniformly distributed over the periphery of the core, but in the case of the alternator the windings are arranged around the armature core as a series of coils, equal in number to the field poles. The spacing of the coils also bear a definite relationship to the pole-pitch of the field magnets, so that the E.M.F.'s induced in each of the series of coils shall simultaneously attain their maximum values. In other words, the changes in the induced E.M.F. in each of the series of coils must be in step or phase.

Present-day alternators may be divided into three classes, viz.

1. Single-phase machines.
2. Two-phase machines.
3. Three-phase machines.

By far the most important are the last two classes.

Referring to Fig. 82, if we have a coil (1) wound in slots *A* and *B*, then the induced E.M.F.'s in the conductors *A* and *B* will be equal and opposite in direction, and the total E.M.F. induced in the coil will be twice that of either conductor. At the instant when the slots are immediately under the poles, the induced E.M.F. will be a maximum, and as the

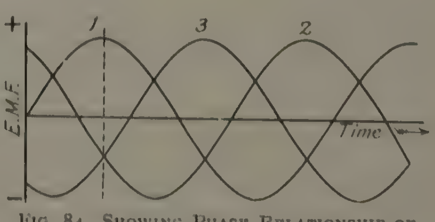


FIG. 84. SHOWING PHASE RELATIONSHIP OF THREE-PHASE ALTERNATING E.M.F.'S

field magnets continue to rotate, the induced E.M.F. in the coil will fall to zero. Further rotation will cause the induced E.M.F. to build up to a maximum of opposite polarity, and again fall to zero, and then build up to a maximum when the slots A and B are again immediately under N. and S. poles respectively. This cycle of changes in the induced E.M.F. is represented by the curve in Fig. 76.

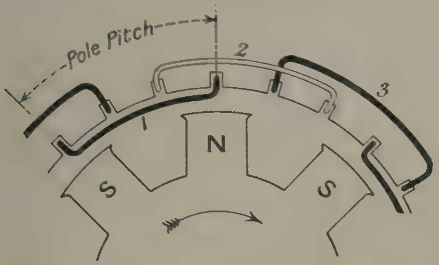


FIG. 85. THREE-PHASE ALTERNATOR WINDINGS

Now let us consider what is happening simultaneously in coil (2), which is angularly displaced from coil (1) by half the pole pitch. The induced E.M.F. in coil (2) will pass through the same cyclic changes as that in coil (1), but at the instant when the E.M.F. in coil (1) is a maximum, the E.M.F. in coil (2) will be zero. Thus, the two E.M.F.'s will have a phase displacement of a quarter of a period, or 90°, as shown in Fig. 83.

If, then, we have a series of coils such as (1) arranged around the stator and connected together to form a single winding, the alternator will be a single-phase machine. By the addition of an independent series of coils displaced relative to the first set by half the pole pitch, the same alternator will be capable of generating two distinct alternating E.M.F.'s, having a phase displacement of 90°. Such an arrangement constitutes a *two-phase* alternator winding. The introduction of this second series of coils utilizes space around the stator core which otherwise would be unused.

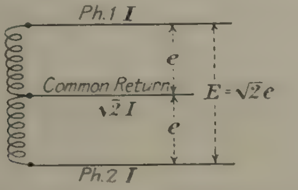


FIG. 86. TWO-PHASE CONNECTIONS

In a three-phase alternator there are three distinct windings, arranged around the stator core so that the E.M.F.'s induced therein have a phase displacement of 120°, as shown in Fig. 84. The spacing of the armature coils of a three-phase alternator is indicated in Fig. 85.

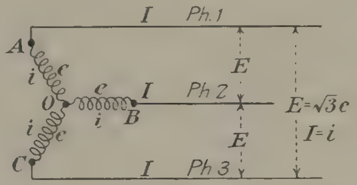


FIG. 87. THREE-PHASE "STAR" CONNECTIONS

TWO- AND THREE-PHASE CONNECTIONS

(*a*) **Two-phase System.** Generally the connections for generators and load on a two-phase system are as shown in Fig. 86, where a common return, or neutral conductor, for both phases is employed. The voltage across the phases is $\sqrt{2}$ times the voltage between each phase and the neutral. The current at any instant in the neutral conductor is $\sqrt{2}$ times the phase current when the system is balanced. The copper section in the neutral conductor must, therefore, be $\sqrt{2}$ times that used for each phase conductor.

In a two-phase common return system, the power is given by the formula—

$$\text{Power} = 2\,E.I.\cos\theta \text{ watts} \qquad (40)$$

Where, E = voltage between phase and common return conductor,
I = line current,
and, $\cos\theta$ = power factor of the system.

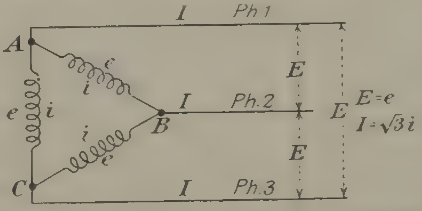


FIG. 88. THREE-PHASE "DELTA" CONNECTIONS

(*b*) **Three-phase System.** It will be seen by reference to Fig. 84 that the algebraical sum of the instantaneous E.M.F.'s in the three phases—and this also applies to the currents in a balanced system—is zero. This really means that the return conductors of each phase can be dispensed with if the three phases are interconnected. Therefore, instead of having six conductors, that is, two per phase, a " go " and " return," it is only necessary to have three conductors, thereby effecting a considerable saving in copper.

There are two general methods of connection for a three-phase system, viz., star (Y) and delta ($\triangle$) connections. The star connection is shown in Fig. 87, and the delta method in Fig. 88. With the star connection, the voltage across any pair of phase conductors is $\sqrt{3}$ times that between any phase and the common point o. The line or phase current is equal to the current in the corresponding arm of the Y connection. When connected in delta fashion, the voltage between the phase conductors is the same as that across the respective sides of the $\triangle$ connection. The line current at any instant, however, is $\sqrt{3}$ times the current in each set of phase windings, when the system is balanced.

The power in a three-phase balanced system, whether star or delta connection, is given by the expression

$$\text{Power} = \sqrt{3}\ E.I.\cos\theta \text{ watts} \quad . \quad . \quad (41)$$

Where, E = the voltage between phases,
I = line current
and, $\cos\theta$ = power factor of the system.

FITTING AND ERECTING

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XII

THE ERECTION OF ENGINES
(*contd.*)

Vertical Engines. These are built in as many variations as the horizontal types. They are single cylinder or compound, the crosshead slides on a single guide face, or between faces, flat, or concave. A single standard may be used with or without a stay bar to support the overhang of the cylinder. In the methods of building up there are many differences. In some small engines, base, standards, and cylinder are cast in one piece. More usual is the casting of the base with the crankshaft bearings, the standards and the cylinder separately. The guides may be cast with the standards, or they may form a separate casting to be bolted on the base, and to be surmounted with the cylinder. Each entails some variation in the methods of adjustment.

The twin standard form is the most common having the guides on opposing faces. The castings will have been machined on the feet, on the top flanges to receive the cylinder, and on the guide surfaces. The points to note in the erection are: the parallelism of the guides, their vertical relations to the base, their right angular relations to the crankshaft bearings, the axial alignment of the cylinder, and the coincidence of its centre with the axis of the guides, and of the crank. The base is levelled, and a commencement made with the crankshaft bearings which are cast with the base. As it is not intended to make any adjustments of these, but to make them the base of operations for the other fittings, the seatings will have been bored, the caps and brasses fitted, bored and faced, and the caps and top half brasses removed. The bearings can be bridged across with a strip of wood on which the centre line will be marked, or a strip across the bore of each bearing can have a central notch, through which a wire can be strained. To either of these the upper parts can be plumbed. The level of the bearings will have been checked when wedging up the base, either with a parallel strip inserted, or a piece of parallel shaft tested with a spirit-level.

From the aspect of facility of erection, the design in which the circular guide is one casting mounted on the base, and the cylinder is mounted on the guides is to be preferred to that of separate standards attached to a base. In the first the parts are faced and registered centrally to each other, in the second some lateral adjustments are necessary. Here the standards are set up first, and the guide faces, plain or concave, aligned from a cord plumbed from the top of the frames to the centre of the crankshaft. The standards being placed in their approximate positions and clamped by their feet, a strip is clamped across the top flanges, through a hole in which the plumb line is dropped to the shaft centre marked on the strip below, or to the strained wire. If the wire is used this affords the slight advantage that the swinging of the line can be arrested by immersing the bob in a vessel of oil below the cross line. Corrections to secure parallelism are made on the feet, or on the facings provided for them, and the surface of the first, or of the second is not continuous, but is bounded with narrow chipping strips. These may not have been planed on the foot at precise right angles with the guiding surfaces, and the erector is able to make minute corrections with the file and scrape—minute, because a trifle removed from the foot will throw the guide surface over by a much larger amount.

The plumb line has to locate the axis of the guides with that of the crankshaft, and with the centre of the crank pin, and the parallelism of the guides throughout their length. These can be tested with a rod gauge, or with internal calipers up and down the guides in the axis. The feet may have to be moved bodily, or corrected to cant the guide faces over in order to bring them into parallel relations. To test their faces in the transverse direction away from the central axis, a thin templet is made to give contact with both guide faces, plain or concave, notched to clear the wire, and is moved up and down. It may be necessary to make some lateral adjustments of the standards. Finally, the bolt holes are reamed and the standards secured to the bed.

The cylinder now has to be aligned with the guides and the crankshaft. It is treated similarly to that on a horizontal bed, with a straining line through the axis. The upper end is bridged with a strip, and a hole is bored through the centre to drop the plumb line through. The cylinder may have to be moved bodily in lateral directions, its axis being parallel with that of the guides but not coincident. Or it may have to be tilted with corrections made on the top flanges of the standards. The shapes of cylinders vary, and the forms of their flanges or feet. They may, or may not, be registered to the standards. They may be open at the lower end with a loose cover, or the end may be solid to include the stuffing box, through which the line will be centred.

After the erector has got the main castings in their correct relations, equal care has to be exercised in the fitting of the moving parts. If this work is poorly done the friction will be serious and somewhat destructive. The detailed parts are prepared separately by squads of fitters in the large establishments, and brought ready to the erector, but in the small shops, he with his helpers may do all the vice work.

The first attention is given to the piston and its immediate connections. A piston comprises its body, cover, and elastic spring rings. A ring in small engines may be in one piece, or in two thicknesses, confined between the body and cover. Or, two, or three Ramsbottom rings may be used in solid pistons. The fitting of these rings is not a minor matter, for on it the efficiency of their action and their durability depend. A ring that makes too tight a fit in its groove, or between the piston and its cover, will be lacking in the expansive quality necessary to maintain a steam-tight joint. A ring too loose will knock itself to pieces. The rings are generally machined very slightly over thickness, leaving to the fitter the task of scraping their faces to make close yet elastic contact. Rings are cut diagonally to prevent scoring the cylinder. After their insertion, the piston should be tried along the whole length of the bore to test its freedom of movement throughout. Many of the larger pistons have helical springs bearing against the internal diameters of spring rings to maintain a measured degree of contact of the rings with the cylinder bore. The piston-rod with a tapered end, or parallel and shouldered, is secured with a nut. The union must be made without any suspicion of slack fitting, which, if present, would cause trouble in working.

The piston-rod cannot be regarded alone, but it must be associated with the crosshead and connecting rod. It is fitted with a parallel, or a tapered end into the boss of the crosshead, and secured with a cotter (Figs. 74 to 76, shown on pages 1062–1065). Afterwards, minute adjustments for length to the crank pin are made in the brasses at the big ends of the connecting rod: in Fig. 77 with a cotter, in Fig. 78 with packings between the flat end of the rod and the brasses. The connection between the crosshead and the connecting rod is made with a parallel pin fitting transversely, Figs. 73 and 75, going through the crosshead and the little end of the rod. The little end has no provision for adjustments lengthwise. It may be bushed solidly, Fig. 77, or have half brasses, Fig. 78. In some cases, the end is embraced by the crosshead, Fig. 73, but generally, the forks of the rod flank the crosshead. The crosshead, the smallest excepted, has slipper plates (Figs. 73 and 74) variously secured, which are adjusted to take up wear and also at the time of fitting up the engine.

In the union of these parts, two essentials are, that the rods shall align, and that the length from the piston faces to the crankshaft brasses shall be correct. If the first are not aligned, then the friction of the crosshead and of the piston and its rod will be in excess on one side, and if the second is not correct there will be trouble with the clearances between the piston faces and the cylinder ends. Some easing of the parts will be necessary, but it must be done in such a way as to preserve alignment.

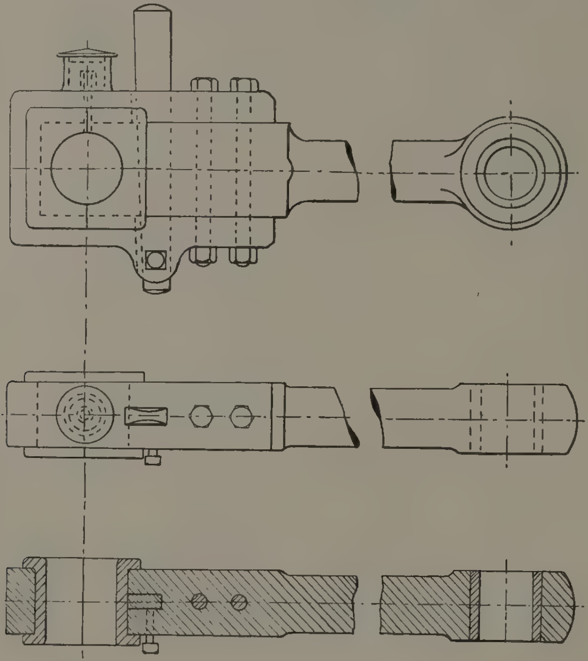


Fig. 77

The piston and connecting rods are packed up on a level table with the crosshead pin vertically. The piston-rod is levelled parallel with the table, and its axial centre taken with a surface gauge. The centre line of the connecting rod is scribed along it, and this has to be levelled with that of the piston-rod. The sides of the small end of the connecting rod will be filed or scraped to a close fit with the crosshead when the two rods are aligned. The amount to be removed will be slight, because a little difference on the faces will be multiplied by the length of the rod. The holes for the pin may have to be scraped a little more on one side than on the other. At the same time, slippers must be checked to ascertain whether they stand out equidistantly from the centre. The gudgeon must be secured according to the method employed.

At this stage the next adjustments are those for getting the precise distance between the piston faces and the crank pin brasses, which are adjusted with a strap and cotter. To determine the exact lengths from piston crank pin the piston is tried in the cylinder with the clearance allowance at each end. It is slight, but necessary, and its amount is stated on the drawings. In vertical engines rather more allowance is made at the lower end than at the upper, but in the horizontal type it is equal. Another matter to note is the bell-mouthing or counter-bore. A spring ring must not go past the parallel-bore, though it may just enter the counter-bore. The diagonal splits in rings must be kept right away from the steam ports, to avoid risk of their opening out there. To make the final adjustments for clearances, the piston is first brought into actual contact with the insides of the cylinder covers. These extreme positions are scribed on the crosshead slipper and on the guide, one corresponding with the other—a permanent record for future reference when readjustments of the brasses are required.

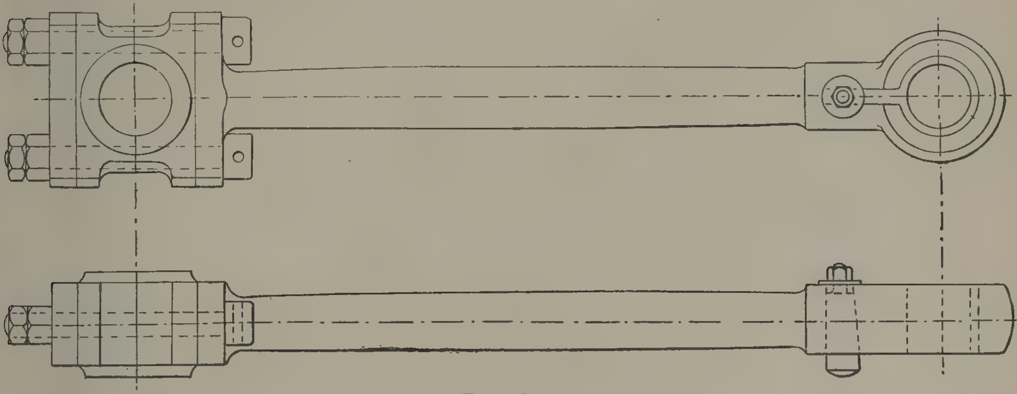


FIG. 78

The valve gear is usually prepared complete from a full-sized drawing, or from a model. If this is not done, the lengths of the link connecting rods have to be trammelled. The valve is set for the lead openings at each end with a thin feeler strip. The eccentrics and their straps will have been keyed on the crankshaft, and the erector has to adjust and connect these to the link motion and the valve rod. There are several minor connections—glands, lubricators, pump, governor, fly-wheel, and packings, all prepared ready for rapid mounting in their places.

OUTLINES OF AUTOMOBILE ENGINEERING

By J. L. Milligan, B.Sc. (Vict.), M.I.A.E.

LESSON VII

IGNITION

In the earliest internal combustion engines the ignition of the mixture was done by a small tube projecting from the cylinder, closed at the end, and maintained at incandescent temperature by a lamp. When the gas was compressed the hot tube ignited it, some control of the ignition position being obtainable by temperature of tube and by the length of the passage from cylinder to tube. Such tubes were mostly of platinum for automobile work, but puddled iron, porcelain, and quartz have been in use.

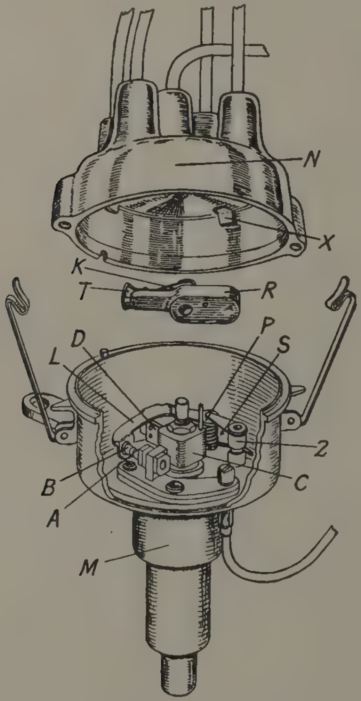


Fig. 32 Delco-Remy Distributor

Electrical ignition is in use in three principal forms. They all involve the use of a sparking plug whose general form is known to every one.

The apparatus must pass an electric spark across a gap of ·015 to 0·25 in. under a gas pressure of 80 lb. per sq. in., and be capable of accurate timing. It must be adjustable in its timing so that the driver has control over the point at which the spark occurs in the stroke of the pistons.

It must spark at speeds as low as 100 rev. per minute of the engine, and as high a speed as the engine runs, say, 3,000 rev. per minute. To produce a spark at the plug under the conditions mentioned requires an E.M.F. of many thousands of volts, and this cannot be obtained direct from a battery or from any simple dynamo. A transformer is therefore used in the circuit. Modern wireless construction has familiarized a very large section of the public with electrical terms, and it is not proposed here to treat of the action of transformers, except to state generally that if two coils of wire are co-axial and a current is passing in one, then any change of current produces an electromotive force at the ends of the other, so that if it forms part of a closed circuit a current will flow; the value of this E.M.F. depends on the rate of change of the current, the number of turns in the two coils, and the nearness of the coils to each other. This was Faraday's discovery. He also found that if a soft iron core was introduced into the coils the effect was very greatly increased. If a few thick turns are used in the primary and a large number of turns (which need only be very fine) in the secondary, then the more rapid the break the higher the secondary voltage.

The break can be obtained by a cam opening and closing a switch, as shown (Fig. 32).

As the current is apt to cause an arc at the switch the break is not dead sharp and hence a condenser is fitted across the break points, which are of a non-oxidizable and hard metal. Platinum, tungsten, and compounds of iridium are generally used. Before dynamos were fitted for lighting, the battery and coil igniter had to be designed to use the smallest possible current in order to give long runs without recharging batteries. This restricted engine speeds very severely.

When the switch of the contact breaker is closed and current begins to flow, the self-induction of the coil and its core of iron wire delays the passage of the energy, and there is a measurable time until the current reaches a maximum value and the maximum magnetic strength of the core is reached. This point is known as "saturation point."

If the engine is a four-cylinder one running 3,000 rev. per minute there are 6,000 explosions per minute, and the period from contact to contact is ·010 sec. It is therefore necessary to have the shortest possible time of *open* circuit on the primary if high speeds are to be attained, unless separate coils and contact breakers are used for each cylinder or pair of cylinders. In order to convey the current to the correct plug a rotating switch, called a distributor, has an arm which comes in turn to contacts leading to each plug, this arm runs at half engine speed. It is usual to combine this with the contact breaker, and put as many cams on the common shaft as there are cylinders, the cam is then a square for four cylinders, a hexagon for six cylinders. As the time for one operation is so small at high speeds, saturation point is not reached and the coil is designed to work below that point. At slow speeds this allows of saturation and, to save the coil from overheating under this excess current, especially if the engine stops on contact, a coil of wire is introduced in the circuit, known as a "ballast" coil; it is made of a metal whose resistance rises rapidly with rise of temperature, this acts as a check on the coil. As an appreciable time occurs between the opening of the contacts on the contact breaker and the passage of the spark at the plug it follows that, if the spark is to occur at the same part of the cycle in the cylinder, contact must be made earlier when the engine runs faster. This is the function of the "ignition advance lever" under the driver's control. The actual *ignition* point needs to be advanced somewhat owing to the time taken to propagate the explosion, but the coil is the chief offender, and many coil sets are fitted with a governor of the ordinary centrifugal type as on gramophones, large steam-engines, etc., which alters the relation of the cam to the shaft it is on. This gives an automatic advance and retard.

If the rear edge of the fly-wheel be polished and smeared with petrol or turps and a wire is brought close to it in some fixed point, then if a spark be passed at this wire instead of at a plug, momentarily, by a two-way switch, a black mark is shown where the petrol was burnt, this enables the lag of the coil to be measured by comparison with the position, when the engine is slowly turned, of the corresponding black spot.

The second type of ignition appliance is to-day only used on one vehicle, the Ford. In this case instead of a battery there are on the fly-wheel a number of permanent steel magnets, horseshoe shape, with their poles outwards. These pass the ends of coils fixed to the engine casing so that the core of the coil is constantly changing its magnetic strength. This produces changes of E.M.F. in the coils which are all connected together, one end being led to a revolving switch in front of the engine whence current is led in turn to each of four transformers or "coils" on the dashboard. The secondaries of these being coupled to the plugs. As the opening and closing of the switch by the engine cannot be made quick enough to give the requisite spark, a second contact breaker or buzzer is fitted on each coil, when the coil is partly saturated, its core draws the "buzzer" blades apart, stopping the flow of current and producing a secondary spark. The core having no circuit to keep it magnetized lets the buzzer return, contact is established, and another spark. There is thus a succession of sparks. This system is simple, cheap, and well understood. No buzzing, no sparks. The gap on the coil contacts has to be adjusted until the right note is found. The switch on the front of the engine camshaft can be rotated in relation of the engine case so as to vary the ignition position.

The next system is the magneto with self-contained coils.

There are two types in use, the rotating coil and the polar inductor. A magneto of the Simms type is shown in Fig. 33. An armature (*a*) rotates on ball bearings between the poles of strong steel permanent magnets which produce a powerful magnetic field. It is wound in two sections, one of which is the primary winding, consisting of a few turns of thick wire; the other being the secondary winding which consists of a large number of turns of fine wire. The rotation of the armature in the magnetic field generates a strong electric current in the primary winding. This current reaches a maximum value twice in each revolution. At this point of maximum value the circuit is interrupted by the opening of the contact breaker which is similar to the contact breaker previously described. The effect of this is to induce a high tension current in the secondary winding, this is conveyed to the insulated ring (*c*) from which it is collected by a carbon brush and conveyed to the distributor (*d*) whence, as before, it is passed to each plug in turn. One end of the primary is led to the body of the magneto and also one end of the secondary. The insulated cover (*e*) carries a carbon stud which rubs on the "live" end of the armature primary and, if the circuit from this stud to the frame or "earth" is closed, there can be no interruption of primary current and no spark. This is the way the magneto is "switched off." As in the coil ignition, a condenser is placed across the switch to give a rapid break of current.

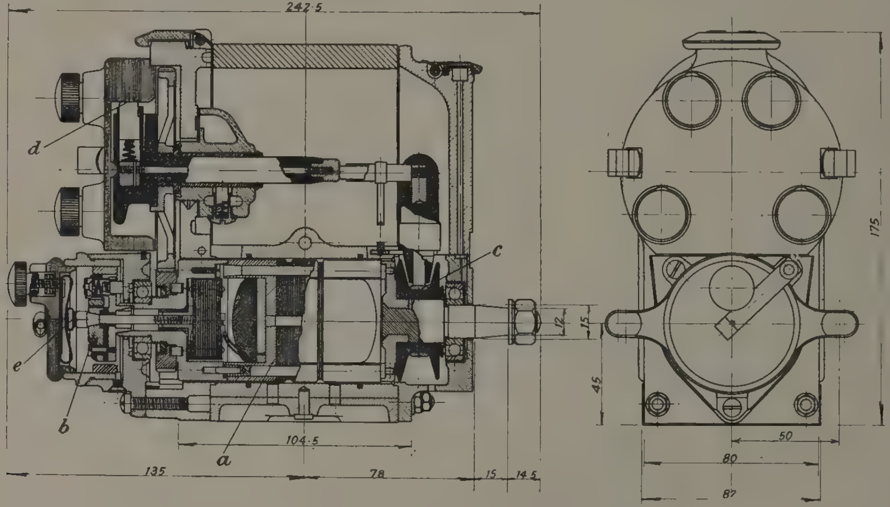


Fig. 33. Section of Simms Magneto Type SR4L

There are many makes of magneto all with substantially the same system of operation differing in detail, but the external dimensions have been standardized to a wide degree to enable interchangeability to be attained, and there is thus a strong family resemblance although the internal arrangements may be different.

As magnetos of this type give two sparks per revolution they run at engine speed for four-cylinder engines, and one and a half times engine speed for six-cylinder engines. The distributor, of course, runs at half engine speed in each case, the gear reduction being therefore 1/2 for a four-cylinder engine and 1/3 for a six-cylindered one.

Order of Firing

Four-cylinder engines fire either 1, 2, 4, 3, or 1, 3, 4, 2. Six-cylinder engines can fire various ways, according to the angle of the cranks, i.e. if No. 2 follows No. 1 or leads it.

A common firing is 1, 5, 3, 6, 2, 4. This does not allow two adjacent cylinders to fire in succession, allowing better distribution in the inlet pipes and exhaust pipe.

The modern magneto is an extraordinary example of the results of patient research in materials and especially in insulation materials. Owing to the high speed of rotation great mechanical stresses are imposed on the windings, also, when a coil is charged the coils tend to come together imposing stresses on the insulation of no mean amount considering the thinness of the enamel, the high temperatures, and the frequency of the stresses. Thin enamel is necessary in order to get the maximum amount of winding on the armature.

The "gap" at the breaker points is of considerable importance, it should not exceed ·016 in. and not be less than ·012 in.; the former figure should be adhered to as closely as possible. The gaps on the two cams should be exactly the same, but it sometimes happens that the cone by which the contact breaker is fitted to the armature is not thoroughly clean when reassembling after adjustment, and the gaps will then vary. The great enemies of magnetos are dirt and water.

Although perfectly watertight magnetos can be purchased, the usual magneto has a few holes to ventilate the machine and prevent "sweating." It is essential that in washing, water be not driven into the machine with a hose, and oil should not be dripped or allowed to creep over it. This rots the insulation and provides a lodgment for dust which forms a conductor. The timing of a magneto is done by rotating the ring carrying the cams. The usual angle is 30° or 35°, the usual setting is such that the break occurs when the piston is at the top point with the magneto cams fully retarded, i.e. moved right forward in the direction of rotation.

The primary voltage rises with the increase in speed and there is not, therefore, so much necessity to vary ignition as in the case of coils. Automatic governors are in use, also impulse starters which can be set at starting so that the magneto shaft is driven by a strong spring, and a catch arranged to stop the magneto shaft about 30° before the firing point. Just at the firing point the catch is automatically tripped and the armature is violently jerked by the spring through the firing position, it is then possible to generate a hot spark with a slow turning of the engine. This is in use on many heavy vehicles.

On lorries, where engines are frequently governed, the ignition is often fixed, also on many omnibus services it is so done for simplicity of control.

RADIATORS

As stated before, if engines were not cooled the temperature would rise to a point where lubrication would become impossible, and no metal would retain its properties.

Small engines and many aeroplane engines are direct air cooled. Automobiles are, with a few exceptions, water cooled.

The cooling is carried out in a radiator, a much misnamed object, which is so designed that it cannot *radiate* much heat. Heat is abstracted by conduction to the air which is drawn through by a fan or by the car's motion. There are three forms, tube, film, and honeycomb. The first consists of two tanks top and bottom with a row of tubes passing from one to the other, either plain or with metal fins threaded on to give a larger surface exposed to the air. This is not a cheap construction, but is used in both large and small cars, expensive and cheap. It is universal among omnibus and lorry manufacturers as it is easy to repair. There are usually three or four rows of tubes, more are worse than useless, as the resulting choking of the air stream more than counteracts the added cooling surface. This is not a light construction as the tubes have to be of substantial gauge, also the fins. The *honeycomb* consists of a number of tubes laid horizontally, each expanded at its ends into either a square or hexagon. These ends are tinned externally, and a stack the size of the radiator front made up in a framing and the ends "dipped" in solder, this tube stack is then fitted into a tank which surrounds it with very little space at the sides. A tank is thus formed, crossed front to back with many air tubes, the water space between them being that provided by the swelling of the tube into a square or hexagon.

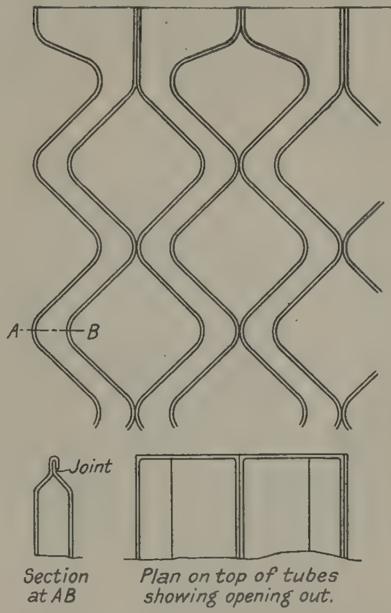


FIG. 34. FILM OR RIBBON RADIATOR

This is a very light construction but costly and very difficult to repair, as well as uncertain in use. The appearance is good, but can be easily imitated on the front tube fins of the tube type. The *film* radiator consists of vertical tubes of peculiar shape (Fig. 34). Let the reader picture two lengths of corrugated sheet (as used for roofs, etc.) standing on edge with ridges horizontal, and the sheets separated by a small amount. The vertical edges are soldered together. There is now a corrugated tank open top and bottom. If another such element is now put touching the first but turned round so that ridge meets ridge, and a third, fourth, etc., all the touching ridges being soldered together. If the top and bottom openings are opened out so that the walls are joined together, this nest of tubes can be fitted with top and bottom tanks and is a "film" radiator. This is not the exact method of construction, but gives a good idea of the system. It is used on a large number of popular makes. Owing to the support from adjacent members and the rigidity due to the corrugated form it is very strong, there is not much soldered joint, and it can be made of very thin material as rolled sheets are used which can be of thinner gauge than tubes.

Radiators are now chiefly made with a false outer shell to take the polish and supply the distinctive appearance, thus masking the radiator itself which can have its joints and construction settled on technical and commercial grounds without sacrifice to the aesthetics of the situation. Incidentally, a "V" or stream-lined front to a radiator throws off air which should go through it, and needs a great deal of tubing to maintain the same efficiency as a flat-fronted tube block.

RECIPROCATING STEAM-ENGINES

By Engineer Lieut.-Commander T. Allen, R.N. (s.r.)

LESSON X

ENGINES FOR USE WITH UTILIZATION OF EXHAUST STEAM

Field of Application. The serious loss represented in condensing steam-engines by the heat rejected to the condenser cooling water, can be avoided or greatly minimized in industries requiring both power and low pressure steam for heating or process purposes. Such cases are to be found in paper mills, bleaching, dyeing and chemical works, tanneries, woolcombing and weaving sheds, sugar refineries, public institutions, hotels, etc. A common practice in such undertakings is to generate power in a condensing prime mover—with accompanying heat loss to the condensing water—and to supply low pressure steam through a reducing valve from the main boilers, or from a battery of low pressure boilers. Occasionally, electrical energy is purchased from an outside source for power purposes, and boilers installed solely to generate the low pressure steam. Such methods imply comparative inefficiency of heat utilization. In the case of the reducing valve, advantage is not taken of the pressure drop from boiler to process steam pressure—the steam being merely throttled through the valve with, as a rule, useless increase of temperature. Where separate boilers are used for the process steam, fuel is burnt to generate steam at low pressure, which, by a small increase in the amount of fuel consumed could be used to produce steam at a pressure suitable for expansion in an engine cylinder.

Principle of Operation. The heat imparted per pound of dry saturated steam at 25 lb. gauge pressure is, from the steam tables, 1,170 B.Th.U.'s, and the total heat per pound only increases to 1,282 B.Th.U.'s if the steam is raised to 180 lb. gauge pressure, superheated 150° F. In other words, the high pressure superheated steam may be produced with an expenditure of fuel less than 10 per cent greater than that required to generate the low pressure saturated steam. It has been shown in Lesson VIII that the steam consumption of an engine is expressed by the relation, $C_a = \frac{2545}{H_d \eta_R}$ lb./i.h.p./hr. where H_d is the adiabatic heat drop from the initial to the final conditions, and η_R is the efficiency ratio.

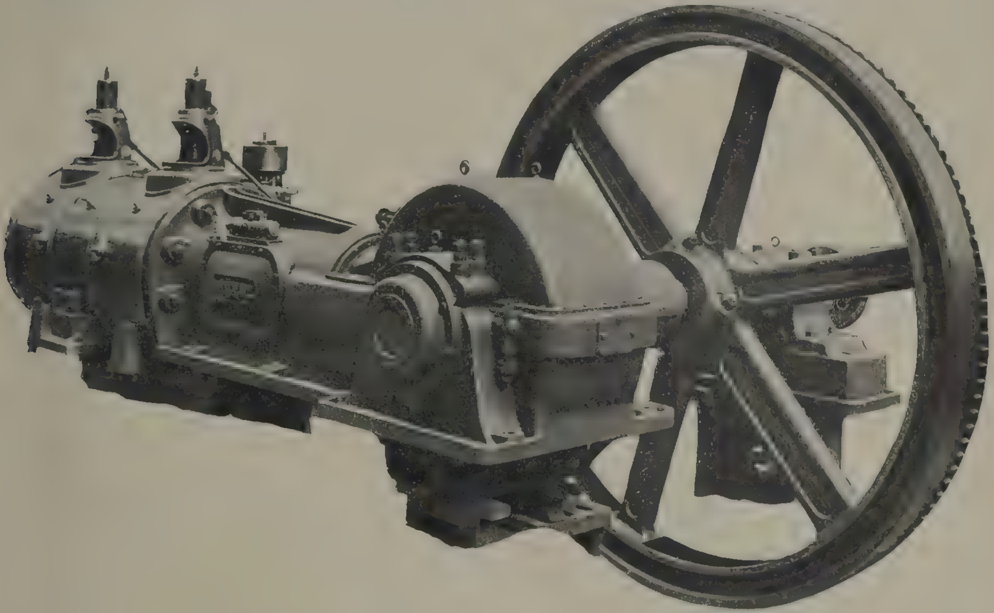


Fig. 32. 150 I.H.P. Musgrave Back Pressure Engine

If, in the above case, it is assumed that the reduction from the high pressure and temperature to the low pressure occurs during expansion in a single-cylinder back pressure engine η_R will be in the region of ·8, and from the Mollier chart $H_d = 140$ B.Th.U.'s per lb. The steam consumption of the engine per i.h.p. hour is, therefore, $C_a = \frac{2545}{140 \times \cdot 8} = 22{\cdot}7$ lb. From this it will be seen that for every 1,000 lb. of steam supplied per hour to such an engine, about 45 i.h.p. may be produced. The steam is exhausted from the cylinder at a pressure of 25 lb. gauge suitable for process use, and both the power and process steam are produced with an expenditure of fuel, less than 10 per cent greater than that required to generate the low pressure steam alone, in a separate battery of boilers. The engine acts, as it were, as a reducing valve; power being produced without cost other than capital charges and operating expenses. This is the principle adopted in back pressure and steam extraction engines, and the source of the large economies possible where such engines are installed under proper working conditions, is apparent. It is obvious from the expression for C_a, that the steam consumption becomes smaller as the heat drop is increased, i.e. as the superheat is increased and the back pressure reduced. It is, therefore, important in such installations to reduce the back pressure to the lowest limit consistent with the proper working of the process machinery, and to utilize a high degree of superheat. The initial steam pressure is fixed by a consideration of the pressure drop required to give satisfactory expansion ratios or cut-offs when exhausting against the desired back pressure.

Control of Back Pressure Engines. It rarely happens in an industrial works that the demand for low pressure steam is exactly met by the quantity of steam supplied to and exhausted from the engine when generating the necessary power. At times of comparative high load and low steam demand, more steam is exhausted than is required in the process machinery and the surplus must, therefore, be exhausted to atmosphere or, in suitable cases, to a steam accumulator. Conversely, if the steam demand is high and the load low, steam must be by-passed from the boilers through a reducing valve into the process main. In correctly designed installations, the atmospheric exhaust and reducing valves are controlled automatically by a pressure governor actuated by the pressure in the process main, which opens one or other of the valves in accordance with the requirements of the moment. The speed of the engine is controlled by a speed governor, which regulates the cut-off to give uniform speed under conditions of varying load or boiler pressure. Where such a dual governing system is applied, the plant is capable of maintaining automatically, constant speed and process steam pressure. Steam exhausted to atmosphere represents a definite thermal loss, and back pressure engines are only justified when, during the greater portion of the time, the steam passing through the engine, at all loads, can be utilized in the process work. In suitable cases it is possible to omit the pressure governing device, and the engine is governed for speed only. The quantity of steam passing through the engine then depends on the load, and the back pressure is not constant. Where a back pressure engine can be connected mechanically or electrically with another prime mover which regulates the speed, the valve gear may be controlled by a pressure governor, so that the load developed by the engine is governed by the demand for process steam, and highly economical working becomes possible. An overspeed device must be fitted to the engine to avoid the possibility of danger in the event of failure of the speed governor of the controlling prime mover. An automatic reducing valve may be fitted to by-pass any deficiency of steam at reduced pressure from the boilers. In all cases, these valves may also be used to by-pass steam at times when the engine is stopped, if the source of hydraulic pressure for working the pressure governor is independent of the engine. Non-return valves should be fitted in the process main to prevent steam passing back from the main to the engine, e.g. when the latter is stopped, and a sluice valve is also fitted, as a rule, close to the non-return valve, for the purpose of isolating the piping when required.

Description of Back Pressure Engines. Fig. 32 is a photograph of a modern horizontal back pressure engine, from which the general features of the design may be observed. The engine is totally enclosed and forced lubricated, and the steam and exhaust valves are operated by eccentrics from a layshaft driven through bevel gears from the crankshaft. The speed governor is visible in the photograph, and in this design the governing forces are transmitted through an oil relay to gear which varies the lift of the double-beat drop valves. Speeds vary between 100 and 200 r.p.m., according to size, and the drive may be direct or through ropes, belts, or gears. Back pressure engines may also take the form of high speed engines with compound or non-compound cylinders. They may be applied either as independent units or in groups exhausting wholly, or in part, to the heating mains or to a condenser or atmosphere, in accordance with the load and heating steam demands of the moment.

Example of Savings. The following comparison between the performance of a horizontal back pressure engine, as shown in Fig. 32, and a condensing engine, with separate generation of the process steam, exemplifies the economies made possible by the first-named prime mover—

Initial steam .	190 lb. gauge pressure, superheated to 575° F.
Normal process steam demand	7,500 lb. per hr. at 5 lb. gauge pressure.
Normal load .	450 i.h.p.

Total live steam required by back pressure engine for power and process steam requirements. = 7,500 lb. per hr.

Live steam required, say, by condensing Unaflow engine, for power requirements only . = 4,800 lb. per hr.
Add live steam equivalent of 7,500 lb. per hr. process steam . . = 6,600 lb. per hr.

Total live steam required for condensing scheme . . . =11,400 lb. per hr.

Saving in favour of back pressure engine = 11,400 − 7,500 = 3,900 lb.-hr., or 34 per cent over the condensing scheme.

FANS AND BLOWERS

THEIR CONSTRUCTION AND OPERATION

By F. G. Whipp, A.M.I.Mech.E., A.M.I.H.V.E.

LESSON VII

PRESSURE FANS

For creating and maintaining pressure above that of the atmosphere, or for the maintenance of a partial vacuum, fans of different design to the propeller are necessary. They may broadly be divided into two classes, viz. peripheral flow and delivery, usually termed *diametrical flow* and *centripetal flow* and *peripheral delivery*, the latter type being the most common in this country. This pattern, too, is often referred to as *mixed flow fans*, but are generally known as *centrifugal fans*.

Diametrical Flow. An excellent example of this type of fan is the "Mortier," a general section of which is given in Fig. 23.

The action of this air-mover, as will be observed from the diagram, is similar to that of a water paddle wheel, the air being scooped up at the circumference of the blades at the inlet passage *A*. It is then taken round with the runner for about a half revolution and delivered into an expanding chimney *B*.

The channel, or by-pass *C*, permits of a certain excess of air being handled, this air being induced by a partial vacuum tendency here and by friction set up by the revolving wheel.

It is possible that a certain percentage of the air drawn in by a fan of this class is carried completely round by the runner, thereby reducing its volumetric efficiency. To prevent this action as much as possible and to ensure delivery at the proper point, the upper portion of the envelope is fitted as near to the periphery of the wheel as practical considerations permit.

It will be evident that the casing is an essential part of the apparatus, for without it there could be no definite flow.

The blades are curved towards the direction of rotation, whereby they tend to scoop the air up when working, and their roots are straightened out so as to incline radially, which permits of the air entering the body of the wheel without serious check.

This pattern of fan does not appear to have found much favour in this country, though it has done and is still doing useful work in some of the Continental mines, mine ventilation appearing to be its chief province.

Centripetal and Peripheral Flow. A mixed flow or centrifugal fan consists of a runner rotating within a suitably shaped housing, usually of spiral formation. Unlike the fan above described, however, it is quite practicable and often desirable for the runner to function without any exterior casing. In such a case the apparatus is known as an "open running centrifugal fan."

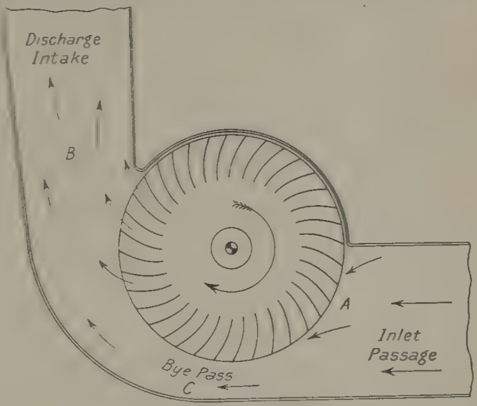


Fig. 23. The "Mortier Fan"

In order to obtain a clear conception of the action of a mixed flow fan, let us assume that we have a bladed runner having two solid side cheeks as in Fig. 24. This runner would also be similar in principle to the "Mortier" wheel, described above. For the purposes of our example there is no casing whatever.

If such a wheel be revolved in free atmosphere there will be a tendency to throw out the air within it owing to centrifugal action, and a slight minus pressure or partial vacuum will be present at the centre of the runner. The throwing out action will occur in all directions radiating from the fan axis, but as no air can enter to fill up the vacuum formed, no flow can take place through the fan. There will probably be some disturbance of the air around the fan periphery caused through friction, and a flow of the external air around the wheel is likely to occur if the speed of the fan is considerable.

If now an orifice be formed in the centre of one or both of the side cheeks, the tendency to form a partial vacuum at the fan centre is overcome by the rush of air passing through these orifices, and a flow results as indicated by arrows in Fig. 25.

Form of Work Done by a Centrifugal Fan. Like the propeller fan, the centrifugal fan does work upon an air stream by increasing its momentum, but it is capable of creating very much greater air speeds than the former apparatus.

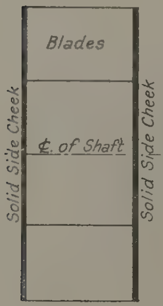


FIG. 24. NO FLOW POSSIBLE WITH THIS RUNNER

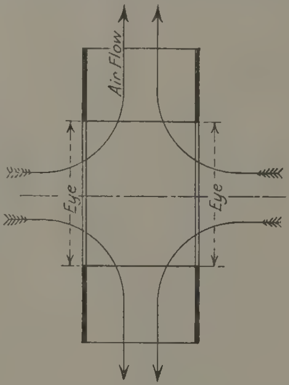


FIG. 25. SAME RUNNER AS FIG. 24 WITH EYES IN SIDE CHEEKS PERMITTING AN AIR FLOW

When the air column enters the eyes—as the orifices in the side cheeks are termed—it is immediately seized upon by the blade roots and thrown outwards towards the circumference of the fan wheel, from whence it is delivered tangentially. Kinetic energy is imparted to the air by reason of its increased momentum, due to the ever increasing product of the radius and tangential component of the absolute velocity of the air.

The speed of the air ejected by a centrifugal fan depends upon the peripheral speed and the form of blades adopted, but primarily upon the former, and it may be established that air speed is a function of the circumferential velocity of the fan runner and, as a correctly proportioned runner should deliver its air over the whole of its breadth at a constant speed, the volume of air delivered is a function of the fan's breadth.

These two factors form the basis of fan design where a given volume of air has to be delivered at a certain speed. It will have been noticed that air speed and not pressure is mentioned as a factor in the proportioning of a fan, the reason being that an open running centrifugal fan working in free atmosphere cannot create a pressure, some external tangible boundary being necessary.

Since kinetic energy is imparted to the air by an open fan, a velocity head must be created, but the whole of this head is lost on delivery, the energy being dissipated in the form of heat.

If, however, the fan be fitted in a chamber which is air-tight, and has only one eye from which it can draw air from an exterior source, then air will be delivered into that chamber until the pressure within it is consistent with the peripheral speed of the fan. When this point is reached no further air will be delivered. This system is often adopted for furnishing forced draught to ships' boiler furnaces where the stoke-holds are specially constructed with air-tight doors. As fresh air is continually being delivered into the hold, no discomfort need be experienced by the stokers; on the other hand, better conditions are often obtainable by adopting this method.

A fan doing duty in this way is virtually delivering into a casing, but as the latter is not designed with a view to assisting in the preservation of the air's energy, many losses will be experienced which prevent the fan from performing its functions with the same efficiency as it would if a correctly designed housing is incorporated.

Purpose of Housing. The object of the ideal housing is to conduct the air delivered into it by the runner to a discharge outlet without shock, friction or eddying, and to convert the kinetic energy of the air into pressure by a gradual reduction of the velocity.

Practical considerations render this ideal unattainable, however, and a well-designed and constructed casing will convert from 60 to 80 per cent of the kinetic energy into pressure head. The housing is assisted in this duty by an external diffuser in the form of a diverging chimney, details of which will be dealt with later.

Conditions Appertaining to Restriction of Fan Inlet and Outlet. If the discharge of the housing be completely blocked up when the fan is working, a pressure equivalent to that of the free atmosphere will exist at the centre of the runner, whilst a somewhat higher pressure will be present at the internal surface of the casing. These conditions are analogous to those obtaining when an open wheel is blowing into an air-tight chamber.

If, however, the order be reversed, viz. the eye closed up and the discharge opened out, the pressure within the housing will be that of the atmosphere, whilst a partial vacuum will be maintained at the fan centre.

When partial restriction of inlet or outlet is made, these conditions will exist in a lesser degree, and they are important to note, for they form the basic principles to be observed in the design of a fan installation. If the fan is exhausting from a system, that is to say, all ductwork is connected up to the inlet eye, an unimpeded discharge must be aimed at, whereas if it is a blowing installation that is being considered, where all the piping is on the discharge side of the fan, the inlet must be free and unhampered if the full benefit of the fan's action is to be obtained.

When the fan has to both exhaust from a piping system on the inlet side and deliver into another plant on the discharge side, the whole installation must be carefully designed with a view to decreasing frictional loss and other forms of resistance to a minimum.

All forms of resistance will be equivalent to a restriction of orifice on whichever side of the fan they occur.

In designing a fan to meet given requirements, its peripheral speed must be such as to give the desired pressure, whilst its breadth must be ample to assure the required air volume being handled. Obviously, the peripheral speed will fix the diameter and speed, and a choice has to be made between a small wheel running at a high rotational speed and a large runner rotating at a slower rate. If the wheel is made too small it may be impossible to adapt an inlet to it which will permit the full volume of air to pass. On the other hand, if it be too narrow, the depth of blades may be excessive and undue frictional losses occur, or construction may become difficult.

Approximate proportions for given patterns of fans have been generally established by the chief fan manufacturers, based on practical observations and tests, but where special conditions of drive have to be considered some deviation from the standard proportions are quite permissible without detriment to the efficiency, but extremes should be avoided.

In standard practice the breadth of a fan runner varies from a third to a half its diameter over blade tips according to the design, whilst the inlet diameter is from ·5 to ·8 of this overall diameter.

The various designs of runners in general use will be dealt with in the next lesson.

WORKS ORGANIZATION AND MANAGEMENT

By E. W. Workman, B.Sc., A.M.I.E.E., etc.

LESSON X

THE STORE-KEEPING DEPARTMENT

Why Stores are Needed. It is impossible to purchase raw material and parts so that they will arrive just when required for the manufacturing processes, and so of necessity purchases must be kept somewhere until required for working on. For this "somewhere" to be a well-organized stores is advantageous from a number of points of view. Stores are essential because—

1. Floor space must be allotted on which to keep all the raw material and parts which are required before manufacture on any large scale can be commenced.

2. Extra storage room is required to enable bulk purchasing to be done, i.e. purchasing of quantities greater than immediate requirements.

3. Stores of material and parts thus awaiting use must be kept together in order that they

may be controlled and hence avoid waste, excessive depreciation, and loss from pilfering or other causes.

4. Properly organized stores are necessary in order to be able to keep the essential records showing what stocks are available for use. It is obvious that without a systematic record of this nature there will be much loss of time, overlapping, and general inefficiency in the handling and recording of materials used.

Functions of Store-keeping. The following are the main functions of a properly organized stores system—

1. *To safeguard material against deterioration, pilfering, and waste.* Deterioration may be due to a variety of causes, such as—

(*a*) Being held in store for too long a time, i.e. purchasing being made too long in advance of requirements.

(*b*) Bad stores accommodation resulting in frequent handling, and the piling of articles one upon another.

(*c*) Dampness, light, and other natural causes which can be guarded against.

Pilfering is best guarded against by devising a stores system which makes it difficult to obtain material except for legitimate and authorized work. The periodic searching of workers as they pass out of the factory is not nearly so effective.

A stores system lessens waste by—

(*a*) Supervising the different receptacles used, seeing that they are the most suitable, do not leak, etc.

(*b*) Ensuring that excessive quantities are not used, and that any surplus material is returned into stock.

(*c*) Drawing attention to materials and articles not being used, so that deterioration and obsolescence may be avoided.

2. *To issue material and parts in an efficient manner and in the correct quantity.* An efficient manner of issuing stores will always mean a saving in cost through high-priced skilled workers not having to waste time waiting to obtain materials. The elimination of this expense will largely depend on the inter-organization of the stores and production departments. If the production department notifies the stores of all material a little before it is wanted, such material can be got ready and delivered by unskilled labour at the place where it will be worked upon and in time to prevent any waiting.

As regards quantity, the exact amount which should be issued is not always known, or the material may be of such a nature that it can only be issued in pieces larger than required for a particular job. In such cases provision must be made for charging out or holding in suspense the charge for the whole amount issued until such time as the surplus not required is returned to stock. In order to ensure that material is only issued legitimately there should be a rule that no material is to be issued without a properly signed requisition as authorization. (This requisition is also required later in the work of the cost department.)

3. *To regulate supply with demand by seeing that stocks are neither too high nor too low.* In order that the regulation of supply with demand may be successful, it is necessary that the store-keeper shall be in touch with probable factory requirements in order that he may always be able to meet any demand and yet keep the smallest possible stock. The lower the value of the stores carried, the smaller is the expense of the capital invested but, on the other hand, work must never be held up for lack of material. For all ordinary stocks the process may be made semi-automatic by deciding for each article the maximum and minimum quantities that may be carried in stock. When the stock records show that the stock of any article has fallen to the minimum quantity decided upon, the store-keeper sends a requisition to the purchasing department for a quantity sufficient to bring the total up to the maximum amount decided upon.

4. *To keep the cost of handling material as low as possible by economy of space and labour.* Expenses can be kept down in connection with heating, lighting, insurance, and wages for handling and issuing material or parts. A quick turnover will also reduce the interest charges for invested capital, and economy of space will allow more area to be given to manufacturing without added rental, and hence reduce the ratio of overhead charges.

Types and Location of Stores. There are four main types of stores that are found in most factories—

1. *The Raw Material Stores.* As this store-room is for the retention of material purchased from outside suppliers, it should be situated so that deliveries may be made easily direct from the conveyances that bring the material. In the case of a large factory, this will mean that both a roadway and a railway line will either enter or run alongside this stores. At the same

time it is essential that the stores shall be conveniently close to that part of the factory where the raw material is first put into use. For this reason it may be advantageous to have the raw material stores divided so that, for example, pig-iron may be stored near the foundry, and wood near the carpenters, etc. Ample room should be provided for the inspection of the incoming material.

2. *The Work-in-Progress Stores.* This stores is situated in the centre of the factory, and is for the storage of completed component parts or semi-completed articles which are held up for some reason or other. The necessity of having such a store is apparent when one thinks of the variety of parts which are needed for an average assembly, and the impossibility of completing all these parts at the same time as they will be individually required.

3. *The Finished Warehouse.* This stores is for the keeping of the finished manufactured articles until such time as they can be packed and dispatched. It will usually be under the control of the selling department, and need not necessarily be situated in the factory at all.

4. *The Tool Stores.* In a large factory a great number of standard and special tools are required, and it will be very conducive to economy in their use if they are properly stored and their whereabouts recorded.

Any of these stores may be subdivided in order to make them more conveniently available to the different manufacturing departments that they supply. In the case of some goods, such as inflammable liquids, it is very necessary to have such subdivision for fire prevention or other reasons. If packing cases are made, they should obviously be stored where they will be required for use. Tool stores should be distributed in the departments where the tools are going to be used. At the same time, in such cases it will usually be found advisable to have an additional central store where the rarer tools may be stored for all departments in order to avoid unnecessary duplication.

Requisitions. Before any material is issued out of a store-room it is essential that the storekeeper should receive an authority to issue in the form of a requisition. The essential information that should be given on the requisition is—

1. Description of the material or part required (or symbol number).
2. Quantity required.
3. The order number to show for what job the material is required.
4. The signature of the person authorizing the issue.
5. Any special particulars such as drawing number or size.

It will usually be found that these requisitions should be received by the stores in duplicate, one copy being retained as proof that the material in question has been issued (and bearing the signature of the person receiving it), and the other copy being forwarded for the completion of the production and cost department records.

Stock Records. These very essential records may be in book form, or, more conveniently, on separate card records for each article and different size. The design of the record will vary with differing requirements, but in any case will have columns which provide for the recording of all incoming and outgoing material according to quantity. It will save duplication of the record in the cost department if money columns are also provided, so that the requisitions can be priced up before being forwarded to the cost department. In addition, columns may also be provided for recording the quantities of material on order, and such particulars as the maximum and minimum quantities authorized to be carried in stock. The cards will be kept in alphabetical and size order, and will be entered up at short and regular periods from the receiving department delivery notes for incoming material, and from a summary of the requisitions dealt with for outgoing material.

METALLURGY FOR ENGINEERS

By James Knowles

LESSON VIII

THE BESSEMER PROCESS—(*contd.*)

The Basic Bessemer Process. Sulphur and phosphorus are not removed under acid conditions and, therefore, pig-iron used for steel making by either acid Bessemer or acid open hearth processes must be free, or nearly free, from these two elements.

Dephosphorization of the iron is impossible in the presence of a highly silicious or acid slag, as the formation of a stable phosphate is thereby prevented. Thus, before any elimination of phosphorus can be successfully undertaken a *basic* slag is a primary essential. This is not practicable except in a neutral or basic lined vessel. Given these two conditions, a phosphate of lime can be formed by which an almost total transference of the phosphorus from the iron to the slag is effected.

In the basic Bessemer process, finally evolved by Thomas and Gilchrist (1877, c.), the formation of the basic slag is ensured by lime additions and the converter lined with burnt dolomite mixed with dehydrated tar. The bottom plate is similarly lined, tuyeres being formed by the insertion of wooden, tapered pins in the perforations during the ramming.

Dolomite is a double carbonate of calcium and magnesium from which carbon dioxide and water are expelled by calcination, leaving a mixture of lime and magnesia. This material has but few binding qualities which deficiency is supplied by the tar.

To revert for a moment to the value of the various elements as fuels for the Bessemer process, the following table should serve as a guide to a comprehension of their respective importance. The initial heat of the molten pig-iron being considered constant, the figures show the approximate increase in that temperature by the oxidation of each 1 per cent of the specified element—

Carbon to carbon monoxide	50° C.
Carbon to carbon dioxide	143° C.
Silicon	188° C.
Manganese	51° C.
Phosphorus (basic)	133° C.

At low (furnace) temperatures silicon has a greater affinity for oxygen than has carbon, but at high temperatures this is reversed. Care must therefore be taken that the silicon is oxidized before the temperature at which carbon has the greater affinity is reached, or the heat will finish with high silicon.

In the basic Bessemer process phosphorus is the principal heat producing element and should be present to an extent of about 2·5 to 3·0 per cent. Consequently, to keep the heat of the bath during the blow within correct limits, silicon should not exceed 1·0 per cent. Manganese usually runs about 1·0 per cent, with sulphur as low as possible.

The plant used is similar in all respects to that of the acid process and, with the exception of the lime additions, the general details of the blow follow closely on the lines previously described. About $3\frac{1}{2}$ cwt. of lime per ton of metal to be blown are charged into the hot converter before running in the iron, and it is important to note that a much greater proportion of slag is produced than in the acid process.

Under working conditions, the removal of the phosphorus cannot be accomplished until decarburization is complete, which stage, it will be remembered, is marked by the drop of the flame. Blowing must accordingly be continued for some few minutes after this and, since there is then no characteristic pyrotechnic display to guide the blower, but only a dense brown fume, samples must be taken from the metal. These are obtained with a long handled spoon after the converter has been turned down and the blast shut off. They are cast in iron chills, hammered down to about $\frac{1}{2}$ in. thick, quenched in water, broken, and the phosphorus estimated from the fracture. Broadly speaking, the more crystalline the fracture the higher is the indicated percentage of this element. If necessary, a further addition of lime is made and the "after blow" continued. Finally, as much slag as possible is poured off the surface of the metal which is then teemed into the casting ladle. The necessary spiegel or ferro is added whilst this operation is proceeding or into the ladle immediately afterwards.

The basic Bessemer process is not worked to any great extent in this country, but is very largely employed on the Continent.

The Swedish Bessemer Process. The Swedish process was the first successful commercial application of the Bessemer principle; the carbon of the steels produced thereby ranges from about 0·15 to 1·50 per cent and, in some respects, its products may be considered serious rivals of crucible steel. There are many points of material difference between it and the English method which may be enumerated and described briefly as follows—

The weight of pig-iron for each blow is much smaller, being only about 3 to 3½ tons, as against 10 to 15 tons in this country and America. This obviously makes for a much shallower bath and consequent reduction in blast pressure, the maximum being about 15 lb. per sq. in.

Formerly Swedish converters were of the fixed type, but these were soon superseded by the tipping or rotating variety. The tuyere area is much greater proportionately, since the *abundance* of the air supply is considered more important than the pressure under which it is admitted, so long as this is great enough to keep the metal out of the tuyeres. This gives rise to somewhat cold heats and, consequently, the converter mouth is contracted slightly so as to conserve as much heat as possible.

The iron is taken direct from charcoal fired blast furnaces worked on cold blast. This is sampled and weighed before pouring into the converter. The blast pressure is adjusted according to the fracture of the sample—the greyer the iron the higher the pressure.

The chemical composition of the iron used is an essential point of difference. This reflects the characteristic purity of Swedish pig-iron—it is very low in phosphorus, contains little or no sulphur, with silicon content at about ·75 to 1·0 per cent. The chief heat producer is manganese, the initial percentage of which is usually about 3 per cent, although this figure may be exceeded as occasion demands. Hence, unless mild steel (0·15 to 0·20 per cent C.) is being made, the presence of this element in such proportions renders additions of spiegel or ferro-manganese unnecessary. Thus, in Swedish practice the formation of ferrous oxide is *prevented*, whereas in the English practice the ferrous oxide is *decomposed*.

The blow is stopped when the colour and sound of the flame indicate a carbon content slightly above that ultimately required. The converter is turned down, blast shut off, and a spoon sample taken which is forged by the smith. The carbon is estimated by several tests amongst which may be mentioned: forging behaviour, bending angle, and magnetic properties. The blow is continued until the correct percentage is obtained.

On account of the relative coldness of Swedish Bessemer heats the percentage of waste tends to run high owing to "skulls" in the ladles. To overcome this a small, specially designed ladle, with nozzle and stopper is employed, and this is bolted on to the converter mouth as soon as it is finally turned down. During the pouring of the steel the bulk of it remains in the converter, the ladle acting as a small reservoir, and so heat radiation losses are reduced to a minimum and solid steel linings in the ladle more or less prevented. The ingot moulds are set on a circular revolving table and brought, one by one, under the nozzle for teeming when a little aluminium is added to each.

A typical example of dead mild Swedish Bessemer steel is that for bicycle tubes, which has a specified carbon content of 0·12 to 0·15 per cent. In this case practically all the manganese is oxidized before the necessary removal of the carbon has been effected, and hence ferro-manganese additions are essential. In steel for this purpose a generous percentage of manganese is permissible, but if a similar temper is required for boiler tubes, since a high manganese tends to increase corrosion, this element must be very carefully controlled. Such steel is therefore made just on the verge of red-shortness, and the blooms are pickled and chipped free from surface defects before working up.

The following are typical compositions of English acid and Swedish Bessemer steels, from which it will be noted that, owing to the higher manganese of the former, the carbon is correspondingly lower—

	English		*Swedish*	
	Dead Mild	Wagon Springs	Dead Mild	Wagon Springs
	%	%	%	%
Carbon . .	·10	·50	·15	·65
Silicon . .	·08	·08	·05	·05
Manganese .	1·00	1·00	·25	·30
Sulphur . .	·06	·06	·02	·02
Phosphorus .	·05	·05	·02	·02

Foundry Bessemer Practice. The manufacture of light castings, for which small quantities of very hot and very fluid steel are required at frequent intervals, owes its tremendous development, in no small measure, to the application of the Bessemer principle to steel foundry practice. The frequent supply of small quantities is achieved by the use of small converters of about 20 to 40 cwt. capacity, and the increased heat of the steel by the introduction of important modifications in the design of these converters.

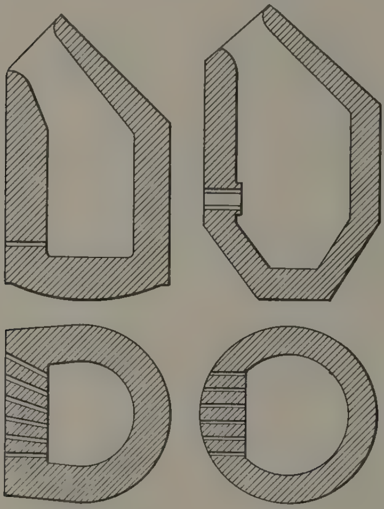


Fig. 9. The Robert Converter

Fig. 10. The Tropenas Converter

Sketches showing essential features of small converters in vertical and horizontal section

In the general description of a Bessemer blow special note was made of the loss of useful heat by the burning of CO to CO_2 outside the vessel; the essential effect of the modifications is to conserve the heat evolved by this reaction *within* the converter. There are two types in use.

The Robert Converter (Fig. 9). In cross-section this is D-shaped, having one flat side in which tuyeres are built horizontally, but set at an angle. The blast, which is admitted at a pressure of 3 to 4 lb., thus imparts a gyratory motion to the metal. The height of the bath is so regulated by the inclination of the converter that the blast impinges on the surface of the metal, or very slightly below it. By these means, unconsumed air is supplied to burn the carbon monoxide formed by the oxidation of the carbon.

Different means are employed to the same end in the **Tropenas Converter** (Fig. 10). This is circular in horizontal section, and as the bottom is in the form of a truncated cone, maximum depth is obtained with maximum surface. The characteristic feature of this converter is the double row of tuyeres, set horizontally and radially in the side. During a blow the vessel is inclined so that the metal reaches exactly to the bottom edge of the lower tuyeres and the blast, of about 3 lb. pressure, skims the very surface. The lower tuyeres only are used during the first stages of the blow, but, on the appearance of the carbon flame, the stopping is removed from the upper series and the oxygen supply thus reinforced at the correct point for the complete combustion of the carbon monoxide.

PIONEERS OF ENGINEERING

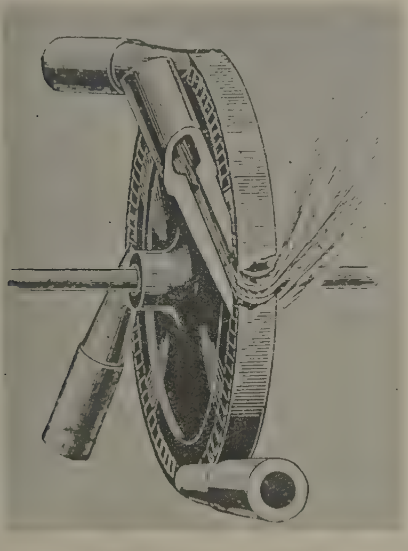
By J. F. Corrigan, M.Sc., A.I.C.

LESSON XXII

THE COMING OF THE TURBINE

SIR CHARLES A. PARSONS—A MODERN PIONEER

The fundamental idea of the steam turbine is as old as the proverbial hills, but the practical application of this early notion is but of comparatively recent accomplishment. Steam-engine experimenters of all ages have constructed various types of experimental turbine engines in which, for the most part, metal vanes were made to rotate by means of a jet of steam impinging on them.

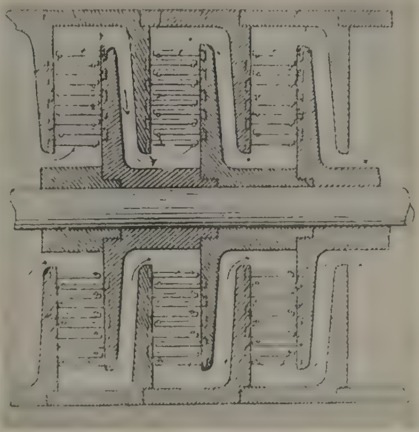


The De Laval Turbine

The first practical steam turbine, however, was constructed by Dr. Gustaf de Laval about the year 1889. In the De Laval turbine the nozzles for supplying steam to the rotor are of the convergent-divergent type.

The great difficulty which the early turbine inventors were not able to overcome lay in the fact that the turbine engine is necessarily a high speed one, and with crude engines working on this principle it was not possible to arrange for a sufficiently high velocity to be given to the working parts in order to extract the greater proportion of the steam's energy of motion. Parsons, however, eventually overcame this difficulty by dividing up the total expansion of the steam into a large number of successive stages. The energy of motion of the steam at each stage of its expansion was thereby reduced, and it became a more practical proposition to devise rotating machinery which would work reliably at a lower speed.

To indulge in a crude explanation—instead of allowing the steam to expand suddenly at one period and to rush with a tremendous velocity through the vanes of a turbine, Parsons contrived to allow the steam to expand in separate and successive stages. Thus, at each stage, the " rush " of the steam was far less in velocity, and its control became much easier.



Section of Early Parsons Turbine

Such is the fundamental principle of the Parsons turbine. In its essentials, the Parsons engine consists of a circular chamber fitted with " guide blades " around its inner circumference. Revolving within this is a large barrel or wheel, also fitted with blades around its circumference. The steam is led through the series of stationary blades, and its force causes the inner barrel or wheel to revolve, the motion being transmitted by suitable means to the main driving shaft of the engine.

The first really practical turbine of this type was introduced about the year 1891, although Parsons had been experimenting with similar engines as far back as 1884. In 1897, the first turbine-propelled vessel—the "Turbinia"—made its maiden voyage. The vessel was merely an experimental one, it being of an order of some 100 tons displacement.

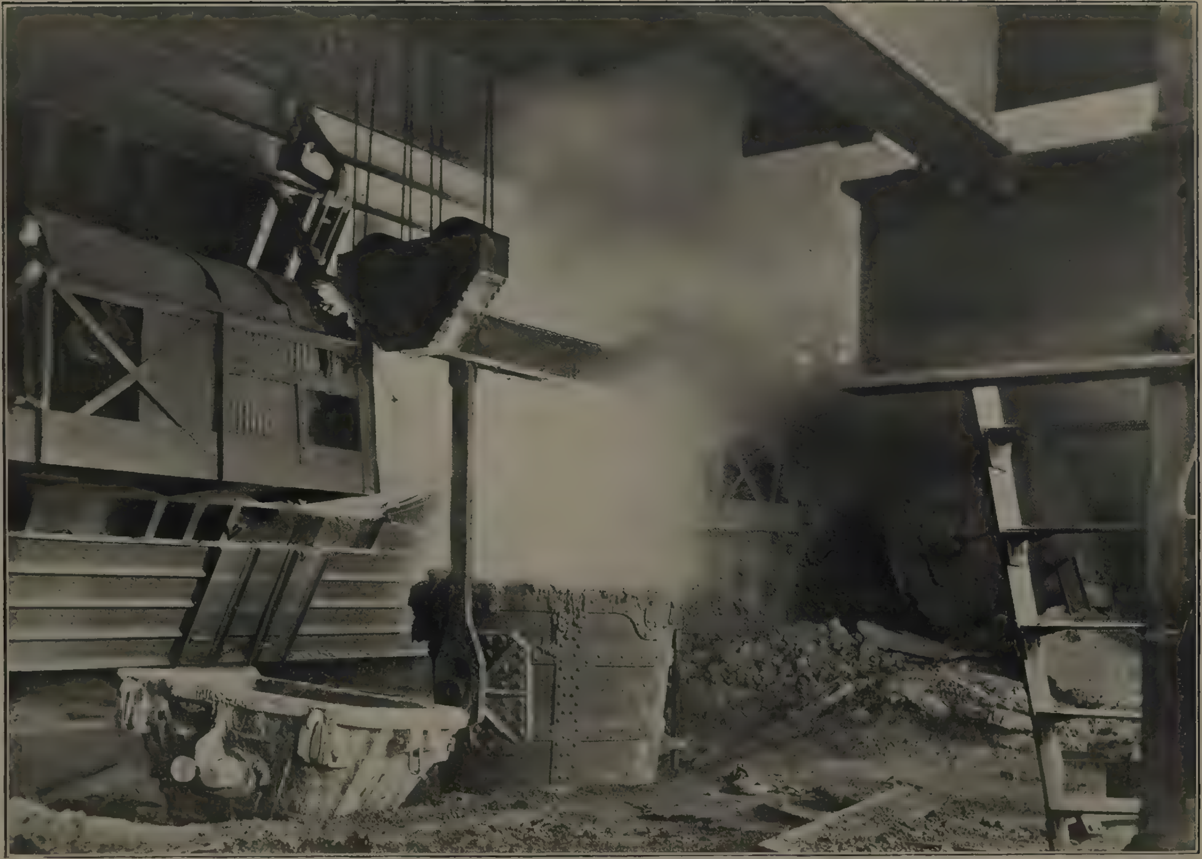
THE "TURBINIA." THE FIRST TURBINE-PROPELLED VESSEL
(From R. N. Neilson's *The Steam Turbine*. Longmans)

In 1904, the Admiralty made a series of trials with turbine equipped cruisers, the "Amethyst" being one which attracted especial attention. As a result of the trials, more naval vessels were equipped with turbines, and, as the reader will be well aware, their use has been still more extended up to the present day, which sees most of the large ocean-going vessels equipped with turbine engines.

The Hon. C. A. Parsons was born in London on 13th June, 1854, he being the fourth son of the third Earl of Rosse, the famous astronomer and constructor of the once celebrated telescope at Birr, Kings Co., Ireland.

After completing his education at St. John's College, Oxford, Parsons entered into partnership with Messrs. Clarke, Chapman & Co. of Gateshead, Newcastle-on-Tyne. This partnership was subsequently dissolved, and in 1899, Parsons built his own works at Heaton, Newcastle-on-Tyne. The firm of C. A. Parsons & Co., Ltd., has now attained universal renown. It is responsible for the construction of various types of stationary turbine engines, but the concern also manufactures dynamos, transformers, searchlight equipment, and other electrical apparatus.

The Hon. C. A. Parsons was elected an F.R.S. in 1898. In 1911 he was created a K.C.B. The work of this modern pioneer of engineering during the late war is well known. As an engineering and electrical expert, he served on various advisory committees which not only dealt with turbine construction, but also with such varied subjects as aircraft, fuel, electric power, and searchlight construction.



TAPPING A LARGE CAPACITY TILTING FURNACE AS EMPLOYED FOR WORKING THE TALBOT PROCESS

(The Cargo Fleet Iron Co.)

METALLURGY FOR ENGINEERS

By James Knowles

LESSON IX

THE OPEN HEARTH

Bessemer's success was followed by a remarkable stimulation of scientific interest in the large scale production of steel, and within the same decade the open-hearth process was evolved. With this the names of Siemens and Martin are inseparably coupled.

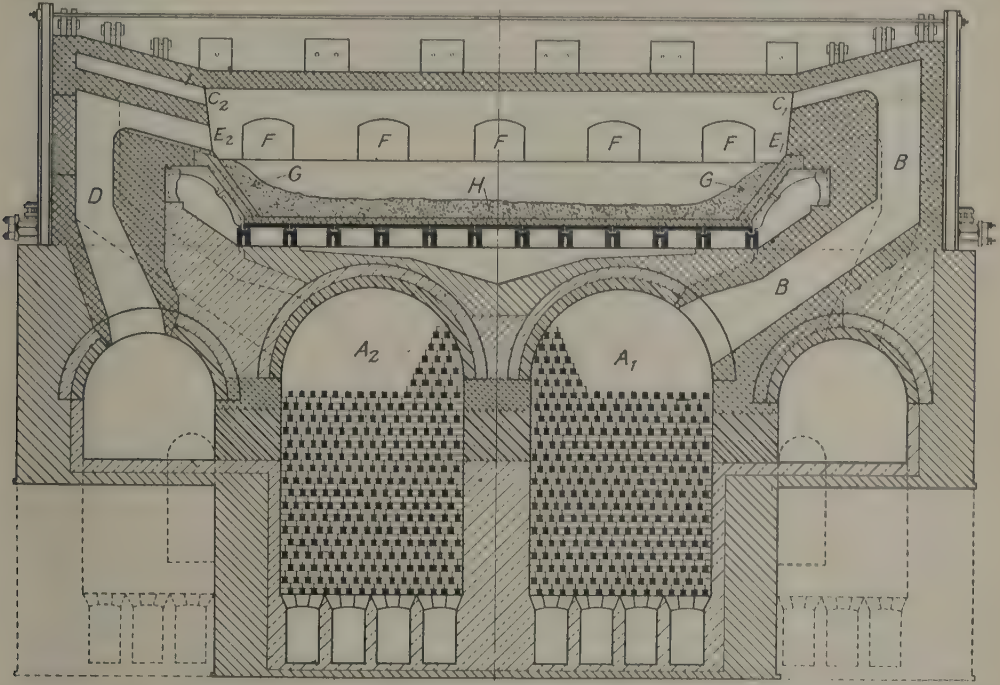


Fig. 11. 60-tons Siemens Open-hearth Furnace—Sectional Elevation

(The Park Gate Iron and Steel Co., Ltd.)

A_1, A_2 = Air regenerators.
B = Air uptake.
C_1, C_2 = Air ports.
D = Gas uptake.
E_1, E_2 = Gas ports.
F = Charging doors.
G = Banks.
H = Hearth.

In France, the latter originated the idea of melting suitable pig in an open sand basin by means of gasified fuel, and diluting the fluid metal by dissolving therein selected wrought iron, or steel, scrap. In other words, since an increase in the amount of metal in the bath was not accompanied by any appreciable increase in the amount of the impurities, the percentage contents of the latter was reduced. This was further assisted by oxidation.

The brothers Siemens, whilst adopting the open hearth, proceeded on the principle of carbon, silicon, and manganese removal by controlled oxidation, using iron ore (Fe_2O_3) as the agent.

A common impediment to the complete development of both these theories, however, was the attainment and maintenance of the intense temperature necessary for working the charge. This difficulty was finally overcome by Mr. (afterwards Sir) Wm. Siemens' invention of his celebrated regenerative system, and, about 1868, the open hearth process was established commercially. A combination of the two methods was ultimately worked, by which the Siemens pig iron and ore, and the Martin pig iron and scrap methods became the Siemens-Martin pig, scrap, and ore process. This was so rapidly developed as to become a serious rival of the Bessemer principle, which, in this country at least, it has now practically superseded.

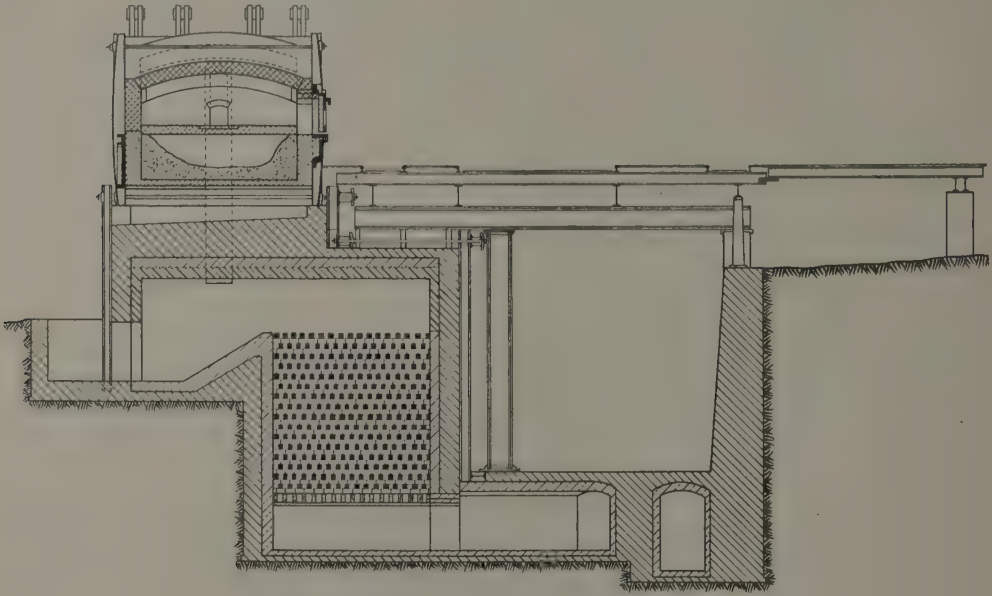


Fig. 12. Section through Gas Regenerator
(The Park Gate Iron and Steel Co., Ltd.)

Dependent on the composition of the hearth, furnace lining, and slag, the open hearth can be employed for either acid or basic practice. Large outputs of special steels, such as nickel, nickel-chrome, and chrome-vanadium are obtained, whilst straight carbon qualities range from 0·10 to 1·50 per cent. Furnace capacities vary from about 5 to 120 tons (50 to 60 tons representing normal present-day practice), the time taken to work a charge being roughly 9 to 14 hours. Several modifications have been introduced, the most noteworthy being the Talbot process, to which later reference will be made. The fuel almost universally used is producer gas, although instances may be quoted where a mixture of coke oven and producer gases has been successfully employed.

THE SIEMENS REGENERATIVE SYSTEM

If the products of combustion in any furnace are permitted to escape unchecked it will be realized that an enormous amount of heat is carried away and lost. Siemens accordingly conceived the idea of recovering, storing, and utilizing this for a preliminary heating of the gaseous fuel and air, thereby materially increasing the heat available for the bath. This he achieved by means of *regenerators*.

These are chambers, usually rectangular in cross-section, almost filled with a chequer work of refractory bricks, generally 9 in. × $4\frac{1}{2}$ in. × 3 in., set on edge and built up in courses so that air spaces and bricks alternate in every direction.

The system is applied to many types of furnaces, and for open-hearth steel melting consists of four units, constructed in pairs, one pair working on each end of the furnace and dealing in turn with the in-going gas and air and outgoing waste gases. Each pair comprises one unit for gas and one for air, separated by a substantial wall to prevent leakage, that for air, as a general rule, being the larger by some 25 to 33 per cent.

The arrangement is such that the gas and air on their way to the furnace and the waste gases going to the stack must first pass through their respective regenerators. The products of combustion, being considerably hotter than the chequers amongst which they circulate, leave their excess heat therein before being finally drawn away by the suction of the chimney. By means of suitable valves the direction of flow is subsequently reversed. The relatively cold gas and air then enter the respective units of these regenerators by inlets situated at the bottom and, absorbing heat, expand. This naturally results in decreased density, increased velocity—or more rapid ascension, and, as a result of further circulation amongst the white-hot brickwork, a progressive gain in temperature. The pressure difference which supplies the motive force is thus constantly accentuated. At the top of the chambers, then, the pressure is sufficient to project the mixture into the furnace in the desired direction, and at such a speed as to neutralize any tendency of the resultant flame to rise to the higher regions of the furnace.

Whilst this is proceeding the opposite sets of chequers are accumulating heat from the waste gases; after a definite interval a further reversal is made, and so the cycle of operations continues, the effect of which may be exemplified by the following figures[1]—

Before combustion—

	Gas	Air
At valves	650° C.	Atmos. Temp.
Leaving regenerators	1,160° C.	1,250° C.
Temperature of combustion	1,803° C.	
Temperature in hearth	1,680° C.	

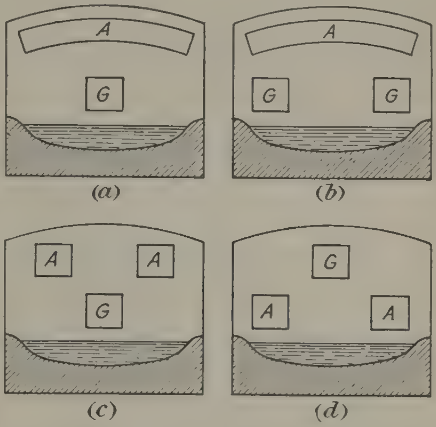
After combustion—

Waste gases leaving bath	1,600° C.
Waste gases leaving regenerators	500° C.

Gases passing through regenerators will obviously take the line of least resistance which indicates the need of very careful design if the maximum transference of heat is to be obtained. Chambers should therefore be vertical rather than horizontal, but, in all calculation and design, the velocity of the gases, the time of contact with and the area of the heating surface exposed are equally essential factors. It is of the greatest importance, too, to note, when considering the weight of chequer work, that the only effective portion of a brick is that within 1 in.[1] of its surface.

THE OPEN-HEARTH FURNACE

The open-hearth furnace, consisting of side walls, end blocks, and arched roof, is built of best quality silica bricks for acid practice. These are stepped at the ends and side to help in forming a basin or hearth. The outer casing is constructed of steel plates carried on girders, the whole being strongly supported by buckstays and tie-bolts. Close to the nuts strong springs are fitted, with which the nuts must be kept in tight engagement, to prevent distortion due to temperature variations. Formerly the regenerators were built immediately under the furnace, which was supported on the top arches, but modern practice is to erect them under the working stage, leaving free air circulation underneath the furnace and obviating the danger of choked up chequers owing to a "break out."



(*a*) (*b*) (*c*) (*d*)

FIG. 13

G = GAS PORT *A* = AIR PORT

The superheated gas and air, on their exit from the top of the regenerators, are conducted through *uptakes* and admitted over the hearth through *ports*. These are inclined downwards so as to direct the flame on to the metal. For good combustion a definite excess of air is necessary and hence the air ports are larger, or more numerous, than those for gas. Again, the volume of the products of combustion is greater than the combined volumes of the in-coming gas and air and, as the same ports have to serve the dual purpose of inlets and exits, they are made larger than is otherwise theoretically necessary. In some American furnaces water-cooled dampers are fitted in the ports affording means of regulating the area to meet these varying requirements.

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[1] These figures have been adapted from *British Siemens Furnace Practice* (F. Clements, M.Inst.C.E. I. & S. Inst., 1922) of which much use has been made in the preparation of this lesson.

Port arrangements (Fig. 13, *a–d*) are both numerous and varied, all having a common purpose, viz., the rapid and intimate mixing of gas and air, in proportions correct for perfect combustion in contact with the metal in the bath. The modern tendency is towards the adoption of a common port, in which the two streams may merge before actually entering the furnace, but in all plants a certain distance (usually about 5 ft.) is left for this to take effect.

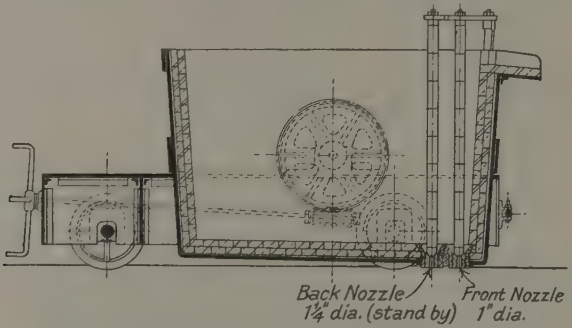

FIG. 14. SECTION THROUGH STEEL CASTING LADLE
(The Park Gate Iron and Steel Co., Ltd.)

The ports are contained in *port* or *end blocks* which form the ends of the furnace and, in view of the heavy demands made upon them, are designed on massive lines. They are often made removable for ease of substitution or repair, are, in some cases, air cooled, whilst for the Talbot process a system of water cooling has been adopted. The distance between the two blocks should be such that the tip of the flame cannot impinge on the opposite end, and complete combustion is effected before the gases make their exit. On the other hand, loss of heat by radiation must be prevented as far as possible. This distance obviously determines the length of the furnace. The width is governed by two main factors: the accessibility of all parts of the hearth for fettling, and the stability of the roof. Formerly, the maximum width was considered to be about 14 ft., but furnaces have latterly been constructed some 16 ft. wide. These two dimensions (length and width), of course, limit the area of the bath. It should be noted that in many large furnaces the roof is not supported by the side walls, but is carried on special girders.

In acid practice the shallow rectangular basin forming the hearth, as already stated, is first given its shape by silica bricks, stepped at the sides and ends (*the banks*). Sometimes crushed gannister or silica rock is rammed over the bottom bricks. A mixture of 95 per cent white silica and 5 per cent red sand is applied in successive layers about ½ in. to 1 in. thick until the desired thickness and contour are obtained, when the bath should drain from every point to the tap hole. Each layer is set hard by the application of heat before the next is added, and whilst fritting will occur between 800 and 1,300° C., at the latter temperature this mixture can be worked into any position. The final layers are composed entirely of white silica sand.

The *tap hole*, about 2½ to 3½ in. diameter, is situated in the centre of the base of that side of the furnace opposite the working stage, and is formed by the insertion of a hard wood, or iron, taper plug before the final layers of sand are put on. Around this ground gannister is well rammed, the plug being withdrawn from outside when the gannister has set.

It is impossible to carry out steel making operations on a bed of pure silica—this must first be saturated with absorbed slag, and therefore a preliminary slag, and sometimes pig, melt is worked, the fluid materials being well rabbled over the banks and hearth. Even then the first few steel heats should be small, and nothing in the way of high grade casts attempted until the hearth is in first-class condition.

The *casting pit* is of such a depth that the tops of the ingot moulds are at approximately ground-level. Above this the *casting ladle* (Fig. 14) travels on rails. Running from the tap hole and forming a channel down which the molten steel flows to the ladle is the *lander*, which may or may not be pivoted. A narrow veranda runs round the tapping side of the furnace to form a working platform and is a continuation of the stage. At the charging side the furnace is provided with several doors, usually 5 or 3, depending on the length, which are raised by hand or hydraulic power. The inlet (mushroom) and reversing (butterfly type, water-cooled) valves are situated below, and operated and controlled from the stage. Each furnace has its own chimney which must be of sufficient height to give satisfactory draught under all conditions.

The general arrangement of an open hearth furnace may be easily followed by means of Figs. 11 and 12, which have been prepared from drawings supplied by The Park Gate Iron and Steel Co., Ltd.

JIGS AND TOOLS

By Leo Kelsey

LESSON X

TOOL-ROOM METHODS OF JIG AND TOOL CONSTRUCTION

Elimination of Fitting. In laying out details of tools which have irregular contours, the method by which they will be made must be kept in mind, as, in the majority of cases, careful manipulation of tools and cutters used in conjunction with the universal milling machine enables the detail to be finished true to form on the machine without further handwork. For this purpose it is desirable that the tool designer should have complete information as to the class and size of all cutting tools used in the tool-room. He must also be familiar with the scope of the universal differential dividing head.

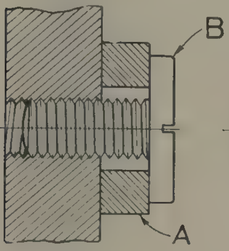

Fig. 87

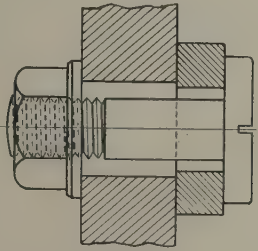

Fig. 88

Unfortunately, space does not permit us to enter into a description of the many uses to which this useful attachment to the milling machine can be put.

Jig and Fixture Boring. When a tool has to bore a number of holes, if necessary, in correct relation to each other within close limits of accuracy, it is essential to provide means of definitely ascertaining the distance between the centres of the holes before they are bored. This is done in the following method by means of small pieces of hardened steel which have been lapped to a definite size. A combination of these in this case is used to mark off, by means of a template or height gauge, etc., the holes to be bored, and drill a hole appreciably smaller than the finished size of the holes required. A disc is now placed over this small hole and clamped in position by the screw *B*, as shown in Fig. 87. The hole in the disc *A* is larger than the diameter of the screw *B*. The work to be bored is now set up on the face-plate of the lathe and a hole bored in it to finished size. Another button is inserted in this hole, with a nut behind it, as indicated in Fig. 88. The next hole to be drilled is now moved into position on the face-plate, and the lathe rotated by hand until the button or disc is running perfectly true. The position between this button and the one with which it has to be in correct relation can now be accurately measured by means of a vernier, or micrometer, over the diameters of the buttons, as indicated in Fig. 89. This method is extensively used on the lathe and the milling machine.

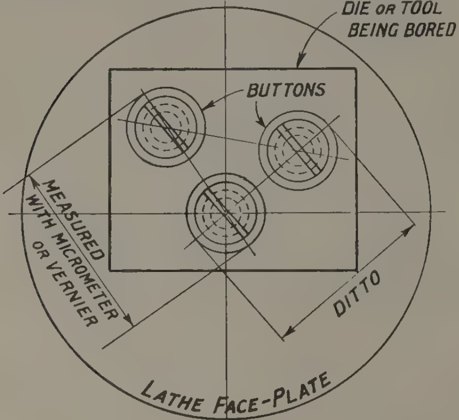

Fig. 89

Use of the Sine Bar. In the measurement of angles, there are two methods in common use: one is the ordinary vernier protractor, which is set to the desired angle within five minutes of a degree by means of the vernier scale; the other method is that known as the sine bar, so called because the sine of the angle, and not the angle itself, is measured.

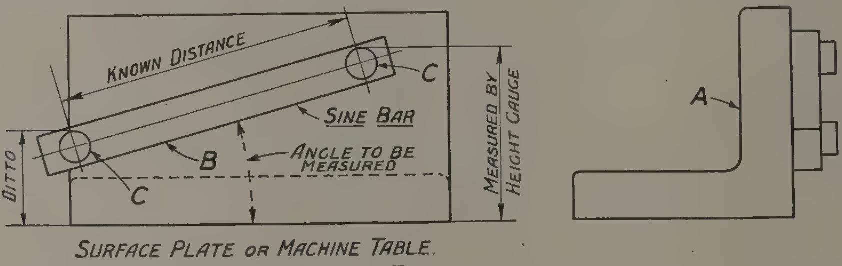

Fig. 90

Referring to Fig. 90, the angle-plate A is perfectly square. Clamped to it is the parallel strip B, to which are attached the discs C. The component is laid on a surface plate, on which the angle-plate A is resting, and is measured by placing it against the parallel B, which is called the sine bar.

The centre distance of the discs C being known, and the difference of their distances from the surface plate, the sine of the angle can be readily worked out. On reference to a table of trigonometrical functions, the angle can then be obtained.

Seasoning. After the iron casting for a tool has been machined and bored ready for insertion of hardened steel bushes (which must be a drive-fit in the former), due partly to expansion of the casting, through the relief of internal strains by machining, it sometimes happens that the holes which have been bored become slightly larger. At the same time, the bushes which have been hardened and ground may become small on their external diameter for the same reason. The result is that, when the two are assembled together, instead of a drive-fit being obtained, merely a push-fit is the result. This has the effect of allowing the bush to rotate inside its housing, finally falling out and becoming useless. Although it is not possible absolutely to control the above conditions, they can be minimized to a large extent by allowing the casting to rest in a cool place on a flat surface for 48 hours, or more if possible, after all machining on it has been finished, and instead of grinding the bushes to a predetermined finished size they should be ground to fit the holes in the casting after the latter have been measured, subsequent to its resting or seasoning process.

Hardening. It is a platitude to say that hardening is the most difficult of the processes in tool-making. Due to the fact that steels, even of the very best quality, will vary slightly in fibre and texture, heat treatment of individual pieces of the same billet will show slightly different results. Of late years, much has been done to reduce the tendency of tool steels to crack and split in hardening, but the trouble is still present, and although it does not primarily concern the tool designer, as he will not be called upon in the majority of cases actually to perform the hardening operation, yet bearing in mind the difficulties which the tool-maker may encounter in hardening tools which he has designed, he must endeavour to eliminate, as far as possible, all thin and slender sections, or large heavy pieces with disproportionately small projections which may be liable to cool more rapidly than the body of the tool, and consequently set up strains which may eventually lead to a crack.

Lapping. This is the process of reducing components to size, using powdered abrasive material. The degree of attrition depends upon the pressure applied and the grade of powder used. A mixture of boiled oil and flour emery, and also rouge, oil, and emery, are those most frequently used. In the case of lapping out drill bushes, a mild steel rod, slightly smaller than the finished size of the bush, is dipped into the lapping mixture, put in the chuck of the drilling machine and rotated inside the drill bush, the spindle being at the same time travelled up and down. This process is repeated until the size is obtained.

Gauge-making Materials. Steels which are to be used for making gauges should be of the best quality, and capable of being hardened to a very high degree, in order that wear may be minimized. Before any work is performed on the gauge, the piece or pieces of steel from which it is to be made should be thoroughly annealed. Unless this is done, cracking is liable to occur when hardening.

Checking Tools. The essential test of all tools lies in their ability to produce the component they were designed for. Their capacity to do this continuously depends on their design, the care with which they were made, and the materials used in their construction. It is not possible here to give a full description of the methods used in checking tools before they are tried out on the machine, but we may state that the procedure is to check the important dimensions first, working back from these until the least important have been measured. Tests should also be made for hardness of locating surfaces, setting blocks, etc., fit of sliding pieces in the case of drill bushes ; fit of slip bushes and diameter of same. Dowel pins should be examined for fit, also all screw threads ; fit of nuts and bolts in the case of clamps, examine clamps to see whether they are seating correctly and not bending. In the case of milling fixtures, machine surfaces should be examined in relation to the base of the fixture, and whether locating surfaces are square with the tenons.

Cutting Tools should be tested by machining components, and by measuring the size or form produced. This especially applies to special milling cutters, drills, reamers, boring bars, counter-sinks, etc.

Although tool checking is not essentially a function of the tool designer, yet he must be able to appreciate the difficulties which the tool-room checker encounters, because the designer is usually called upon in the case of variation from drawing dimensions to adjudicate between the tool-room and the tool checker, who is usually a member of the inspection department of the factory.

(Conclusion)

THE THEORY OF HEAT ENGINES

By E. B. Cole, B.Sc.

LESSON XIII

FLOW THROUGH NOZZLES—(*contd.*)

Critical Pressure. In considering adiabatic frictionless resisted flow of steam through a nozzle, we had just become familiar with the idea that a *critical* back pressure exists. If the back pressure be higher than this critical pressure, we have seen that the nozzle will be convergent and the mass flow will be determined by the *exit* area of the nozzle. If, however, we expand down to a pressure below the critical, the nozzle first converges and then diverges, and the mass flow is determined entirely by the *throat area* of the nozzle.

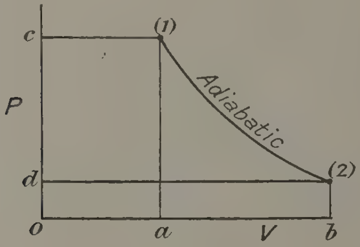

Fig. 40

Hence, it is very important to determine this critical pressure.

Now we have proved that

$$\frac{v^2}{2gJ} = I_1 - I_2 \text{ heat units per lb.}$$

and, while we have dealt specifically with steam, this equation is true for any substance, and we have seen in the case of steam how easily the solution may be obtained from the T–ϕ or ϕ–I charts for this substance.

Let us now examine this equation in a general manner for any substance.

We have

$$I = E + \frac{PV}{J} \textit{ heat units,}$$

where V is the volume occupied by the substance, or, measuring I and E in *ft.-lb.*, we may write

$$I = E + PV \text{ ft.-lb.}$$

$$\therefore \frac{v^2}{2g} \text{ ft.-lb. per lb.} = (E_1 - E_2) + (P_1V_1 - P_2V_2)$$

Now for *adiabatic expansion*, as represented by the PV diagram, Fig. 40, the work done, which is given by the area $a12b$, is entirely performed at the expense of the *internal energy*—no heat being supplied or rejected.

Thus, area $a12b = E_1 - E_2$ ft.-lb. But

$$\text{Area } oc1a = P_1V_1$$
$$\text{Area } od2b = P_2V_2.$$

$$\therefore \frac{v^2}{2g} = \text{area } a12b + oc1a - od2b$$
$$= \text{area } c12d$$

= area of indicator diagram for admission, adiabatic expansion, and exhaust for the substance.

For steam, we have already seen that this is approximately the indicator diagram for the Rankine cycle, thus the result has been proved in another way.

Assuming that the equation of the expansion curve is $PV^n = c$, where n is the index of adiabatic expansion, we have already proved that the area under such an expansion curve is

$$\frac{P_1V_1 - P_2V_2}{n - 1} \text{ ft.-lb.}$$

$$\text{Thus, area } c12d = \frac{v^2}{2g} = \frac{P_1V_1 - P_2V_2}{n-1} + (P_1V_1 - P_2V_2)$$

$$= \underline{\frac{n}{n-1}(P_1V_1 - P_2V_2)}$$

(a result the student was asked to deduce in a problem in the section on the gas laws).

This result is true for any substance, where n is the *adiabatic* index.

Re-arranging this expression, we have

$$\frac{v^2}{2g} = \frac{n}{n-1} \cdot P_1V_1 \left\{ 1 - \frac{P_2V_2}{P_1V_1} \right\}$$

$$= \frac{n}{n-1} \cdot P_1V_1 \left\{ 1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \right\}$$

since $P_1V_1^n = P_2V_2^n$,

$$\text{Thus, } v = \sqrt{\frac{2g \cdot n}{n-1} \cdot P_1V_1 \left\{ 1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \right\}} \cdot (1)$$

Now, if V_1 and V_2 represent the *actual* volumes per lb. at (1) and (2),

Then the mass discharged per second will be—

$$w = \frac{Av}{V_2}, \text{ where } A = \text{area in sq. ft.}$$

$$= \frac{A}{V_2} \cdot \sqrt{\frac{2gn}{n-1} \cdot P_1V_1 \left\{ 1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \right\}}$$

V_2 can be expressed in terms of the initial known conditions—

$$P_1V_1^n = P_2V_2^n \therefore V_2 = V_1 \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$$

$\therefore w =$

$$= \frac{A}{V_1} \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \sqrt{\frac{2gn}{n-1} \cdot P_1V_1 \left\{ 1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \right\}}$$

$$= A \sqrt{\frac{2gn}{n-1} \cdot \frac{P_1}{V_1} \left\{ \left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}} \right\}} \quad (2)$$

Now the initial conditions fix P_1 and V_1, thus A, n, P_1, V_1 are all constant, so that the mass flow w depends on the ratio P_2/P_1. If we write r for this ratio, then

w is a maximum when $r^{\frac{2}{n}} - r^{\frac{n+1}{n}}$ is a maximum, which from consideration of the calculus will occur when

$$\frac{d}{dr}\left(r^{\frac{2}{n}} - r^{\frac{n+1}{n}} \right) = 0$$

$$\text{i.e. } \frac{2}{n} \cdot r^{\frac{2}{n} - 1} - \frac{n+1}{n} \cdot r^{\frac{1}{n}} = 0$$

$$\text{i.e. } \frac{2}{n}\, r^{\frac{1-n}{n}} = \frac{n+1}{n}$$

$$\text{or } \quad r = \left(\frac{n+1}{2}\right)^{\frac{n}{1-n}} = \underline{\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}}$$

This expression gives us the ratio of the back pressure P_2 to the initial pressure P_1 at which the discharge is a maximum, hence knowing the value of n we can at once determine the critical (throat) pressure.

If the expanding substance is a perfect gas, from the relationships already proved we have

$$n = \gamma \text{ (i.e. ratio of } K_p/K_v).$$

For *Air* $\gamma = 1{\cdot}41$.

$\therefore$ substituting above

$$r = 0{\cdot}527$$

i.e. reducing the back pressure below 0·527 × (initial pressure) does not increase the mass flow any further.

Returning to steam as the expanding substance, we have also seen how a mean value of n for adiabatic expansion may be determined.

Dr. Zeuner, however, gives an equation by means of which the value of n may be obtained for steam of given initial dryness—

$$n = 1{\cdot}035 + 0{\cdot}1x$$

where x is the initial dryness fraction.

Thus for steam initially dry

$$n = 1{\cdot}135$$

$$\text{and } \therefore \quad r = 0{\cdot}577.$$

If the value of n for different initial dryness fractions be obtained, it will be found that, over the range likely to occur in practice, the ratio P_2/P_1 will not differ materially from 0·58, and this value may, therefore, be taken for all cases of saturated steam.

For *superheated* steam, which remains superheated throughout the adiabatic expansion, the value of n is 1·3, so that $r = 0{\cdot}546$.

Maximum Discharge of Saturated Steam. Considering again steam *initially dry and saturated*, and substituting 0·577 for P_2/P_1 and 1·135 for n in equation (2), we obtain an expression for the *maximum discharge* of steam through the given area A, which will be the throat area—

$$w = 3{\cdot}60\, A \sqrt{\frac{P_1}{V_1}} \textit{ lb. per second}$$

where

A = area in sq. ft.

P_1 = initial pressure in lb.-sq. ft.

V_1 initial volume, here equal to volume of 1 lb. dry saturated steam at pressure P_1.

Maximum Discharge of Superheated Steam. If the steam remain superheated throughout the whole of its expansion as above, substituting $n = 1{\cdot}3$ and $r = 0{\cdot}546$ in equation (2), we have

$$w = 3{\cdot}786\, A \sqrt{\frac{P_1}{V_1}} \text{ lb. per second,}$$

where V_1, the initial volume, is the volume of 1 lb. superheated steam at pressure P_1.

In the *Steam Turbine* section of this EDUCATOR these results will again be stressed and amplified.

Let us, in conclusion, work out the following two examples—

PROBLEMS. (1) Determine the throat and exit dimensions of a nozzle to pass 1,800 lb. of AIR per hour, assuming adiabatic frictionless flow, under the following conditions—

Initial pressure P_1 = 200 lb. sq. in. abs.
,, temperature T_1 = 3000° F. abs.
Exit pressure = atmospheric, 15 lb. sq. in.

We are here dealing with a "perfect" gas and can, therefore, use the gas laws to determine the unknowns.

We have then for air,

$$\text{Critical throat pressure } P_2 = {\cdot}527\,P_1 = 105{\cdot}4 \text{ lb. sq. in.}$$

(The suffix 2 will be used to denote the back pressure concerned, here—the throat conditions.)

The temperature of throat T_2 is given by—

$$\frac{T_1}{T_2} = \left[\frac{P_1}{P_2}\right]^{\frac{\gamma-1}{\gamma}} = \left[\frac{1}{0{\cdot}527}\right]^{\frac{{\cdot}41}{1{\cdot}41}} = 1{\cdot}205$$

$$\therefore \underline{T_2 = 2490^\circ \text{ F. abs.}}$$

The velocity at throat v_T is given by

$$\frac{v_T^2}{2g} = \frac{\gamma}{\gamma-1}[P_1V_1 - P_2V_2] \text{ ft.-lb. per lb.}$$

This expression can, for a gas, be conveniently simplified to

$$\frac{v_T^2}{2g} = J.K_p(T_1 - T_2) \text{ ft.-lb. per lb.}$$

since $PV = RT$
$\gamma = K_p/K_v$
$R = J.(K_p - K_v)$ ft.-lb., the specific heats being measured in heat units, K_p for air being 0·238.

$$\therefore v_T = \sqrt{2g \times 778 \times 0{\cdot}238\,(3000 - 2490)} = \underline{2460 \text{ ft.-sec.}}$$

where g = 32 ft.-sec. per sec.

From $PV = RT$, the volume of air at throat is given by

$$V_2 = \frac{53{\cdot}2 \times 2490}{105{\cdot}4 \times 144} = 8{\cdot}74 \text{ cub. ft. per lb.}$$

$$\text{Mass flow per second} = \frac{1800}{60 \times 60} = {\cdot}5 \text{ lb.}$$

$$\therefore \text{Area at throat (sq. in.)} = \frac{8{\cdot}74 \times {\cdot}5}{2460} \times 144 = \underline{0{\cdot}255 \text{ sq. in.}}$$

$$\therefore \text{Diameter at throat} = \underline{0{\cdot}57 \text{ in.}}$$

In exactly the same way, expanding from the initial conditions right down to the exit pressure of 15 lb. sq. in. we have, using suffix 2 to represent now the exit conditions—

$$T_2 = 3000 \times \left[\frac{15}{200}\right]^{\frac{{\cdot}41}{1{\cdot}41}} = \underline{1410^\circ \text{ F. abs.}}$$

$$v_2 = \sqrt{2g \times 778 \times {\cdot}238(3000 - 1410)} = \underline{4330 \text{ ft.-sec.}}$$

$$\text{Volume per lb. } V_2 = \frac{53{\cdot}2 \times 1410}{15 \times 144} = \underline{34{\cdot}7 \text{ cub. ft.}}$$

$$\therefore \text{Area} = 144 \times \frac{34{\cdot}7 \times {\cdot}5}{4330} = \underline{0{\cdot}576 \text{ sq. in.}}$$

2. Calculate the throat and exit diameters of a nozzle to discharge 3,600 lb. of STEAM per hour under the following conditions—

Initial pressure = 200 lb. per sq. in. abs.
Exit pressure = 1 lb.
Initial dryness fraction = ·97.

Thus, the critical throat pressure = ·58 × 200 = $\underline{116 \text{ lb. sq. in.}}$

Consulting steam tables, we have

1. *Initial* Absolute temperature "corresponding" to 200 lb. per sq. in. abs. = $\underline{843^\circ \text{ F.}}$
Latent heat = $\underline{843{\cdot}3 \text{ B.Th.U.}}$

2. *Throat* Absolute temperature corresponding to 116 lb. per sq. in. abs. = 799° F.
Latent heat = 880 B.Th.U.
Specific volume = $\underline{3{\cdot}44 \text{ cub. ft.}}$

3. *Exit* Absolute temperature corresponding to 1 lb. per sq. in. abs. = 562° F.
Latent heat = 1035 B.Th.U.
Specific volume = $\underline{333 \text{ cub. ft.}}$

The following results have been obtained by calculation of the Rankine heat drop, and the student is advised to work them out in detail—

(*a*) *Throat* dryness fraction = ·93
Rankine heat drop = 43 B.Th.U.

$$\therefore \text{Velocity at throat} = \sqrt{2g \times 778 \times 43} = \underline{1465 \text{ ft.-sec.}}$$

$$\therefore \text{Area at throat} = \frac{3600 \times {\cdot}93\,(3{\cdot}44)}{60 \times 60 \times 1465} = {\cdot}002185 \text{ sq. ft.}$$

$$\therefore \text{Diameter} = \underline{0{\cdot}63 \text{ in.}}$$

(*b*) *Exit* dryness = ·746

Rankine heat drop, from initial pressure, 200 lb. to 1 lb. = $\underline{324 \text{ B.Th.U.}}$

$$\therefore \text{Velocity at exit} = \sqrt{2g \times 778 \times 324} = \underline{4020 \text{ ft. sec.}}$$

$$\therefore \text{Area at exit} = \frac{3600}{60 \times 60} \times \frac{{\cdot}746\,(333)}{4020} = {\cdot}0618 \text{ sq. ft.}$$

$$\therefore \text{Diameter} = \underline{3{\cdot}37 \text{ in.}}$$

(CONCLUSION)

OUTLINES OF LOCOMOTIVE ENGINEERING

By A. Morton Bell, O.B.E., M.I.Mech.E., M.I.Loco.E.

LESSON V

AUXILIARY APPARATUS—(*contd.*)

An important detail of a locomotive, which seriously affects its efficient and economical working, is the draft derived from the exhaust steam escaping from the blast pipe. Innumerable efforts have been made, dating from the very first locomotives, to regulate and make variable this draft. Unfortunately, nearly all arrangements necessitate the use of mechanical apparatus in the smoke-box, where it is subject to the destructive action of the gases of combustion, exhaust steam, etc. Only limited success can therefore be credited to most of the schemes, although many different ones can be found in considerable use. Perhaps, one of the most simple and practical arrangements is the provision of a means for splitting, or dividing, the cone of steam as it escapes from the blast nozzle, causing it to present a ragged surface to the gases it is required to exhaust and expel up the chimney.

Other favourite forms of adjustable blast pipes consist of expanding cones, movable caps, and mechanically-operated valves, whereby a portion of the escaping steam can be diverted from the blast nozzle, should it be deemed desirable to reduce the draft.

On superheater locomotives it is often customary to provide a pyrometer, as it is found that a record of the temperature of the superheated steam enables the fireman better to control his fire, as also the driver to have better knowledge of the work his engine is doing. Such refinements, however, are usually confined to locomotives employed on long and continuous runs.

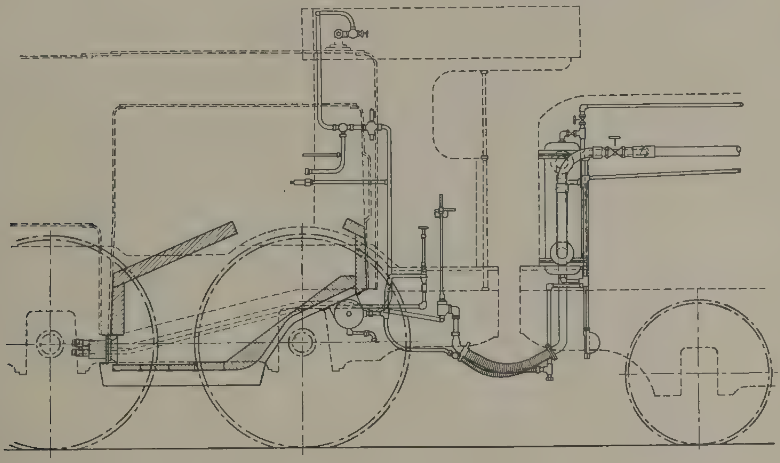

Fig. 16a. Oil Fuel Arrangement

In many parts of the world where coal is scarce or expensive and oil easily procured, liquid fuel is much employed. In this country it has only received somewhat limited application when controlling circumstances have suggested and permitted its use. The drawing (Fig. 16a) shows an application on a modern system wherein the oil fuel is preheated after leaving the tender tank, and then injected, or sprayed, into the specially built furnace arranged in the fire-box for ensuring its complete combustion. The sprayer, or burner, actuated by steam, is placed at the bottom of the fire-box, so that the whole of the cubical contents of the latter are available, as it is essential that the oil should be perfectly consumed therein, and only heated gas enter the tubes.

Attempts have been made to spray and distribute the oil fuel by pressure alone, as it is recognized that the introduction of steam into the fire-box for this purpose must result in a considerable loss of heat units, for although the steam may be ultimately disintegrated into its ultimate gases O and H, the heat required to be absorbed in the splitting-up process is more than that which will be returned by any future combustion. Such arrangements have been adopted with great economy and success in stationary power plants and on board ship, but there are great difficulties in providing the pumping plant and other details required, on a locomotive where space is so limited, so that spraying by steam, which should certainly be superheated, remains the more practical method.

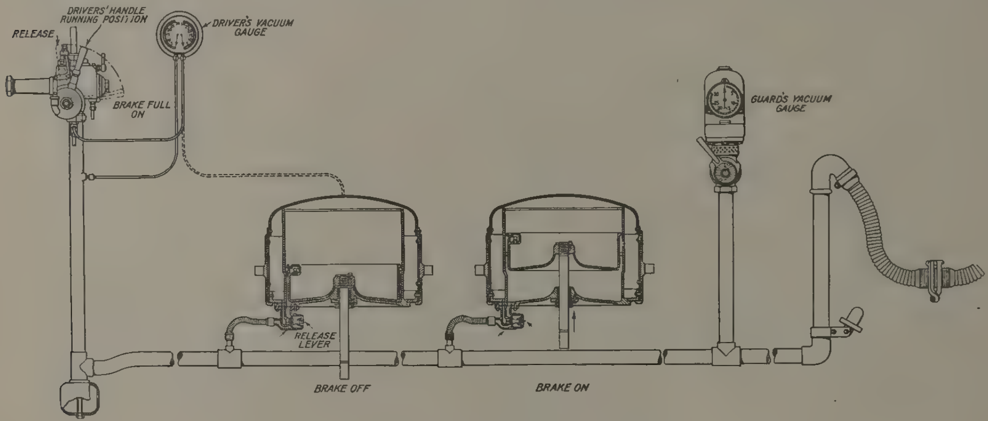

Fig. 17. Vacuum Automatic Brake

Another important auxiliary apparatus is that required for operating the continuous brakes of the trains. There are two systems in use in this country—compressed air and vacuum—the latter being the more generally used on passenger trains of the steam railways, whilst the former is almost exclusively employed on electrically operated and super-rapid steam suburban lines. The vacuum brake depends on the partial exhaustion of the apparatus, which is then subject to atmospheric pressure for its power. The exhaustion of the brake pipes, cylinders, chambers, etc., is secured by pumps and ejectors. In all cases, vacuum is first created by a steam ejector, but the continuous vacuum to be maintained is more economically secured by a pump operated by the mechanism of the engine.

The drawing, Fig. 17, shows the apparatus in detail, and the following brief description will enable its operation to be readily understood. The ejector on the engine creates a partial vacuum throughout the system, the cylinders under the vehicles being exhausted both above and below their pistons, through the ball valves. The pistons being in equilibrium fall by gravity, and the brakes remain off, but on atmospheric air being admitted to the train pipe, either through operation of the driver's valve by desire, or through an accidental severance of the pipe, it closes the ball valves against the upper side of the cylinders, and air enters and exerts pressure on the lower side, causing the pistons to be pushed up and apply the brakes. Re-creation of vacuum and exhaustion of the spaces above and below restores the equilibrium, and the pistons fall, releasing the brakes. As stated, the vacuum can be maintained either by a pump or by a small steam jet, incorporated in the large ejector.

Owing to the fact that it is necessary for the brake cylinders operated by atmospheric pressure to be of very large diameter to secure the requisite power, there is often difficulty in locating such cumbersome details among the closely packed parts of a modern locomotive. Many engines are therefore at the present time equipped with either steam or air brakes for the engine, working in conjunction with vacuum ejectors and apparatus for the brakes of the train.

Fig. 18 shows a tank engine whereon air

brake apparatus is provided for the engine, as well as a vacuum ejector and appliances for operating the brakes of the train. A special valve on the footplate ensures both systems of brake acting simultaneously and synchronizing when applied or released. The diagram, Fig. 18A, shows these fittings.

In the case of air brakes working on the well-known Westinghouse principle, the power for operating the brake apparatus is compressed air, a supply of which is maintained throughout the train by a steam pump located on the locomotive. A pressure of 75 lb. per sq. in. is maintained in the train pipe, but the main reservoir on the engine is charged to 90 lb. per sq. in. Under all the vehicles equipped, an auxiliary reservoir is provided with a brake cylinder, wherein the movement of the piston is controlled by a triple valve.

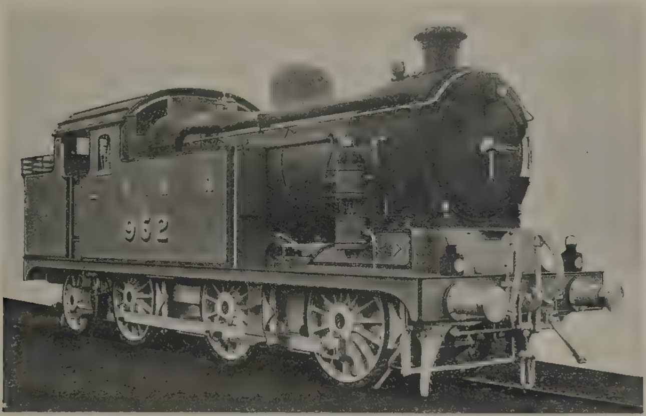

FIG. 18. LOCOMOTIVE WITH AIR BRAKE AND VACUUM APPARATUS FOR TRAIN BRAKES
(Robert Stephenson & Co., Ltd.)

In operation, the compressed air, after passing from the main reservoir on the engine through the driver's valve, flows along the train pipe, lifting the triple valves of each vehicle, then charging the auxiliary reservoirs. Under these conditions, the brakes are "off." When the air in the train pipe is allowed to escape, either by design—the driver or guard opening the valves provided—or by accidental severance of the pipes, etc., the pressure will be reduced and the air stored in the auxiliary reservoirs exerting its superior pressure, will move the triple valves into such a position that the stored air pressure will pass to the brake cylinders, forcing the pistons out and applying the brakes.

The brakes are released by again restoring the pressure throughout the train pipe, by admitting air from the main reservoir on the engine (maintained at higher pressure for rapid action). This will cause the triple valves to lift, allowing the air in the brake cylinders to escape to the atmosphere, and thus relieve the brake blocks from pressure. At the same time, the triple valves open up communication again between the train pipe and the auxiliary reservoirs, to enable them to be recharged to the running pressure.

The elementary arrangement described has been very much improved in late years by the Westinghouse Brake Co., and with the refinements they now recommend, practically instantaneous action, both in application and release, can be assured for trains of almost any length.

For the successful braking of very long goods trains, it would appear essential that some form of high pressure apparatus must be adopted, on account of the length of time which elapses in the restoration of low pressure, or vacuum, throughout the piping and apparatus of the vehicles forming such trains.

There is another phase of this brake operation by pressure which might be mentioned here, and that is the great advantage secured, when operating with compressed air, of having a supply of power on the locomotive equipped, available to operate the reversing gear, sanding apparatus, rocking grate, and many other auxiliaries which are rapidly becoming too heavy and difficult to operate by manual effort.

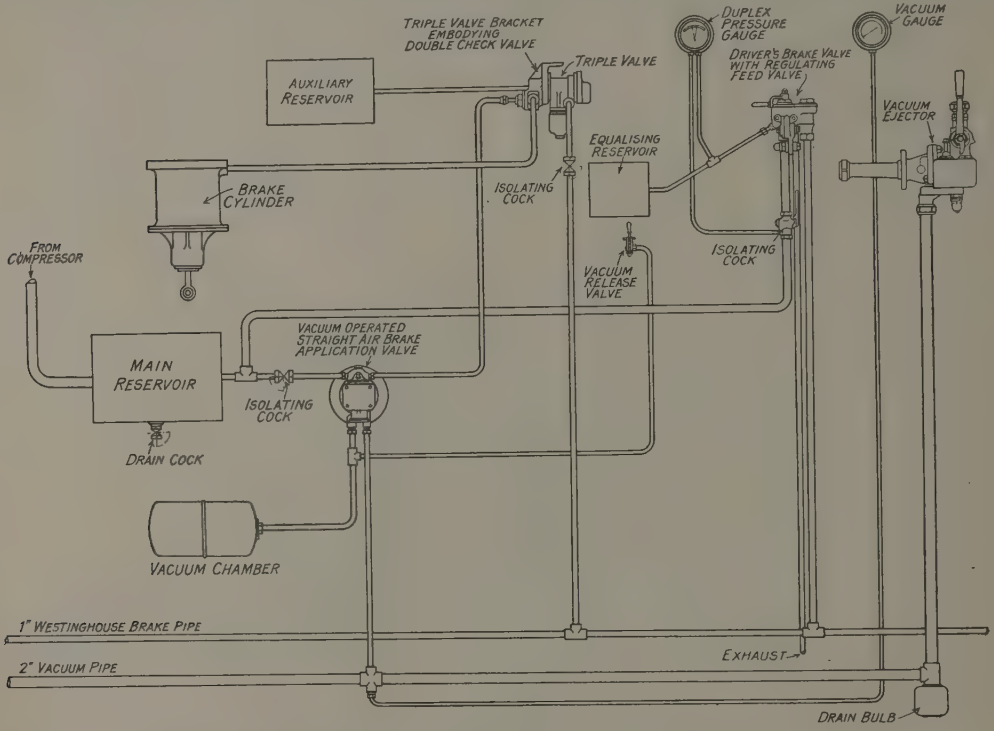

Fig. 18a. Diagrammatic Arrangement for Operating Westinghouse Air Brake in Conjunction with Vacuum Brake

Electric lighting apparatus is now frequently included in the equipment of locomotives intended for service on Colonial and foreign railways traversing long stretches of sparsely settled country. A small steam turbine driven dynamo, or generator, is located on the boiler, and the current taken from it to a switchboard in the cab, whence it is distributed for lighting up any of the lamps required for headlights, signalling, or inspection purposes. There are instances where short trains, consisting of a carriage or two, are lighted up by this same installation, but it is only to be recommended for secondary services.

THE TESTING OF PRIME MOVERS

By R. Royds, M.Sc., M.I.Mech.E.

LESSON VI

THE MEASUREMENT OF BRAKE HORSE-POWER

The brake horse-power of an engine refers to the power developed or given out at the crankshaft. The mechanical efficiency of the engine is given by the ratio $\frac{\text{b.h.p.}}{\text{i.h.p.}}$, but in the case of a pump, compressor, or similar plant, where a.h.p. is the power required to drive the pump, then the mechanical efficiency is $\frac{\text{i.h.p.}}{\text{a.h.p.}}$

The brake horse-power of an engine is usually measured by means of some suitable form of "brake." For the sake of demonstration, suppose winding drums are fixed on the engine shaft of radii R and r and running in the direction shown in Fig. 30, these radii being measured to the centre lines of the ropes wrapped round the drums. The weight W is ascending and the weight w is descending; the weights of the ropes are neglected in the present calculations. In one revolution made at constant speed the work done on W is $2\pi RW$, and that done by w is $2\pi rw$. Thus the net work done in one revolution is $2\pi(WR - wr)$, that is, 2π × net moment of forces about the centre; and no matter how many additional drums with their ropes and weights we might imagine, the same expression would result. If N were the revolutions per minute, with the loads measured in pounds and radii in feet, then,

$$\text{Brake horse-power} = \frac{2\pi \times (\text{net moment, lb., ft.}) \times N}{33{,}000}$$

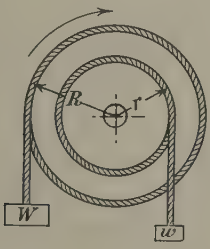

Fig. 30. Winding Engine

If no motion of the weights took place and slippage between the ropes and the drums allowed the weights to be suspended in the positions shown, the net moment resulting would be the same as before. Therefore, whatever type of brake may be adopted, what is required from the brake is a measure of the moment of the forces acting on the brake, for at constant speed that must be the moment acting on the shaft.

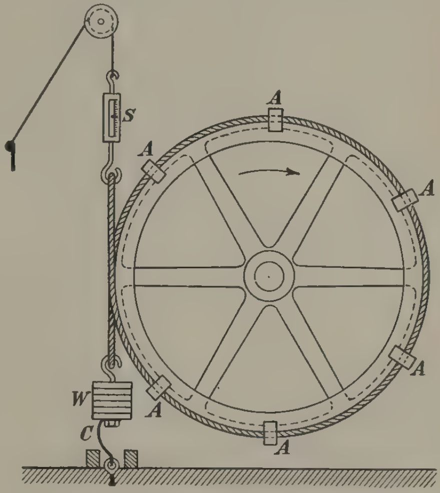

Fig. 31. Rope Brake

The Rope Brake. A common arrangement of the rope brake is shown in Fig. 31. The rope makes one turn round the brake wheel, supporting at one end the weights W, the other end being connected to the spring balance S. A cotton rope is preferable because of its flexibility, and it is better to steep it in a mixture of melted tallow and black lead to exclude moisture rather than to use it in the dry condition. The rope may be $\frac{1}{2}$ in. diameter for small engines and, say, $\frac{3}{4}$ in. or 1 in. diameter for engines of 50 b.h.p. running at moderate speeds. Preferably the rope would be put on the wheel double so as to form a loop for attachment to the hanger at *W* and also at the spring balance end, and where the loops pass one another, the loop to *W* should be accommodated inside the loop passing to *S*. To space the ropes on the wheel and to keep them in place, wooden blocks *A* are wired to the ropes, cut as shown in the sectional view in Fig. 32. A loose anchor rope or chain *C* may be used to safeguard against the weights being carried over the top should the frictional force between the rope and wheel increase suddenly. This increase is sometimes caused by careless handling of the rope at the spring balance when reading the balance or when adjusting the weights to a floating position.

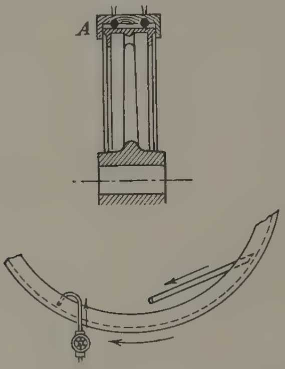

Fig. 32. Water-cooled Brake Wheel

For tests at moderate loads and of short duration the rope brake may be applied to the smooth rim of the fly-wheel, but otherwise it is preferable to use a water-cooled brake wheel with the rim of channel section. On a plain fly-wheel the heat developed at the rim is likely to cause expansion of the rim, while the arms try to prevent this expansion. This might produce a dangerous condition in a cast-iron fly-wheel, and, even if nothing worse happens, is liable to cause the fracture of one or more of the arms. The brake wheel shown in Fig. 32 would be suitable for diameters greater than, say, 3 or 4 ft. For a small wheel it is a common practice to cast it solid with the boss instead of using arms, with the channel section rim offset on the outside of the wheel. Fig. 32 gives a sectional view through the type of wheel shown in Fig. 31, as well as showing how the cooling water may be supplied and "picked up" from the inside of the channel. The water supplied to the channel immediately spreads out with the wheel running, due to centrifugal action. The heated water can be picked up by a pipe cut at an angle and set at a suitable slope as shown, with the cut end of the pipe nearly touching the rim on the opposite side from that on which the water is supplied. If the rim has a central web, this web has to be cut away at a few places round the rim.

The rope type of brake is not very suitable for high speeds of rotation, with the correspondingly small brake wheel, except at very small powers, because difficulty is experienced in keeping the rim cool when heavy loads are attempted, owing to the small cooling surface. Canvas, tape, or chair webbing is generally more satisfactory than ropes on small brake wheels.

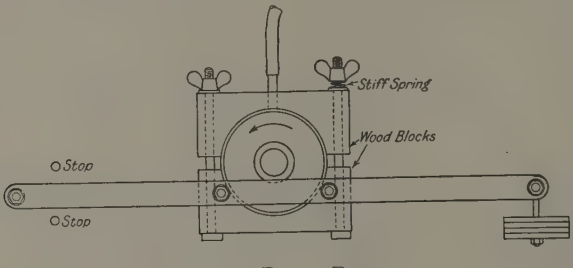

Fig. 33. Prony Brake

A rope brake sometimes has a tendency to get into a state of oscillation, especially if used for testing a reciprocating engine. These oscillations can usually be damped out by introducing frictional resistance into the spring balance. In bad cases it may be found necessary to jam the spring balance, releasing it only at the moment a reading has to be taken.

In estimating the weight W at the heavy end of the rope, the weight of the hanging rope from where it leaves the wheel should be added to the load on the rope. Similarly, with the arrangement shown in Fig. 31, the weight of rope to the point of contact with the wheel should be deducted from the spring balance pull to give the pull w at the light end of the rope. If these loads are in pounds and the radius R is measured to the centre of the ropes, then

$$\text{Net moment} = (W - w)R \text{ lb. ft.}$$

$$\text{and b.h.p.} = \frac{2\pi(W - w)RN}{33{,}000}$$

where N is the revolutions per minute.

The Prony type of brake shown in Fig. 33 is capable of absorbing more power than a rope brake when applied to small brake wheels running at high speeds, though sometimes the Prony brake is not very manageable. In this arrangement two blocks of wood are cut to fit the wheel as shown, and are pulled together by two bolts to create frictional resistance between the wheel and the blocks. Arms are bolted to one of the blocks and the load is applied to the brake at some convenient place on the arms, either by means of weights or by a spring balance. A copious supply of water is necessary to absorb the heat developed at the rim and to preserve something like constant frictional resistance. This water is best applied at the point shown by a flexible connection to a water main, and grooves cut in the blocks as in a bearing will help to distribute the water over the surface of the wheel. The water is thrown off from the sides of the rim, and, unless the arrangement is for temporary use and the water thrown off is not objectionable, sheet iron casings can be fixed to the blocks and the excess water collected from a hole in the bottom block. Even then it is difficult to prevent water from being thrown about. A stiff spring under the wing nut of one of the bolts helps in the adjustment of the brake to the conditions desired. If the load W lb. is applied at the distance R ft. from the centre, then

$$\text{Net moment} = W \times R \text{ lb. ft.}$$

$$\text{and b.h.p.} = \frac{2\pi WRN}{33{,}000}$$

With tests of long duration the practical limit of this type of brake on small brake wheels at high speeds is about 20 to 25 b.h.p. per per sq. ft. of rim contact.

A type of band brake suitable for slow speed engines up to about 500 b.h.p. is shown in Fig. 34. Two semicircular steel bands, $\frac{3}{16}$ in. thick, are faced with pieces of cotton belting, and are pulled together at the ends so as to create frictional resistance between the brake wheel and the band. The load on the brake is partly due to the weights hanging from the outer end of the arms connected to the brake bands, and partly due to the pull on the spring balance acting on similar arms on the other side. Ordinary bolts may be used to connect the two semicircular parts of the brake together, but it will be found more convenient to introduce some arrangement on one side capable of fine adjustments. A toggle gear arrangement, which is displaced by a right- and left-hand screw, is

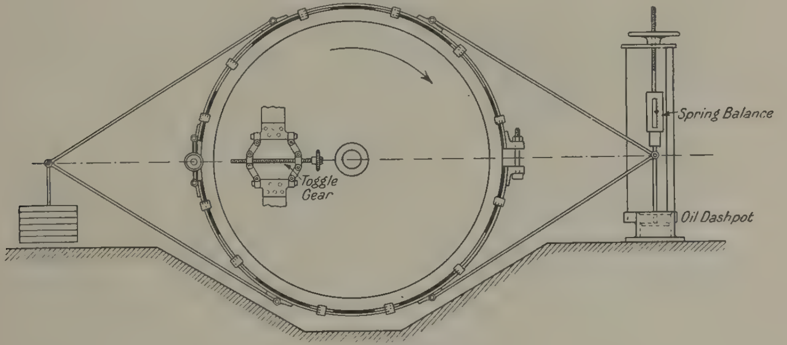

Fig. 34. Band Brake

very suitable for this purpose, and is shown in Fig. 34. An oil dashpot would probably be found to be necessary to damp out the tendency of a brake of this type to oscillate as the engine rotates. This dashpot is shown in Fig. 34 as a circular plate working in oil in the cylinder below the spring balance. This plate should *not* be made a fit in the cylinder, but left quite free. If there is any tendency to oscillate the oil offers a resistance to the rapid motion of the brake, but does not interfere with slow motions such as those required to keep the brake in adjustment. A channel section rim bolted to the fly-wheel of the engine, and cooled by water in the manner shown in Fig. 32, forms a serviceable brake wheel. With a fairly copious supply of water this type of brake will absorb from 10 to 14 b.h.p. per sq. ft. of cooling surface. The working surface of the rim and the pieces of cotton belting should be kept free from dirt, rust, oil, or water, and should be treated occasionally with graphite or blacklead to give something like constant frictional resistance at heavy loads.

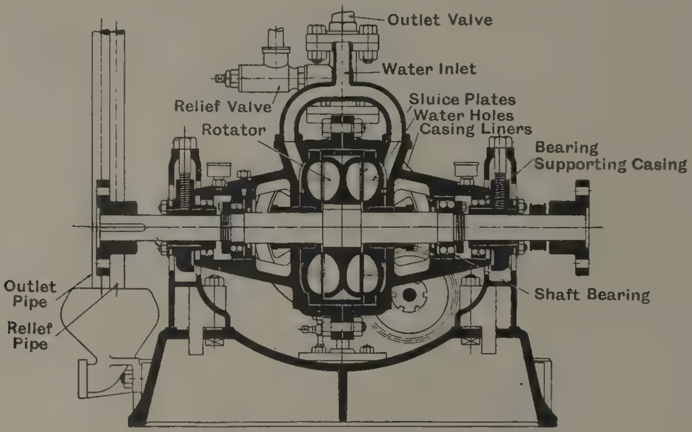

FIG. 35. FROUDE HYDRAULIC DYNAMOMETER

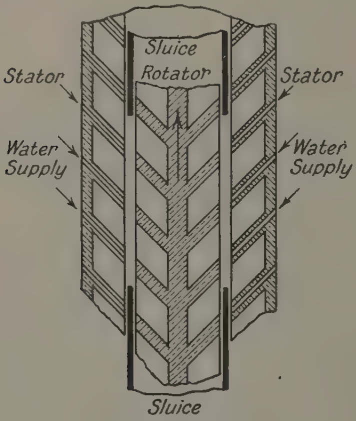

FIG. 36. SECTION THROUGH POCKETS OF FROUDE DYNAMOMETER

Different forms of this type of band brake have been used, some with wooden blocks in place of the cotton belting here recommended. Wood blocks do not work well on cast-iron rims unless kept thoroughly wet, because the frictional resistance between the rim and dry wood is usually very erratic when the brake heats up.

Fluid Friction Brakes are sometimes used for the absorption and measurement of small or moderate powers at high speeds of rotation. In the simplest form it consists of one or more circular plates or discs fixed to the shaft and surrounded by a casing so that the discs rotate in water. The rotation of the casing is prevented by a known load applied at a definite distance from the centre of the shaft. The moment on the shaft can only be varied independently of the speed by varying the quantity of water in the brake, but this quantity is not easily adjusted to steady conditions, as a constant flow of water is necessary to absorb the heat. The brake may have a tendency to erratic action under such conditions. The moment of resistance for a single disc running completely drowned is

$$\text{Moment}^1 = 100 \times f \times N^2 \times R^5 \text{ ft.-lb.}$$

where N = rev. per min.
R = radius of disc, ft.
f = 0·002 to 0·003 for plain metal discs.

This expression would hold approximately for each disc on the shaft, provided that division walls on the casing intervened between each pair of discs.

The Froude Hydraulic Dynamometer or brake, built by Messrs. Heenan & Froude, Ltd., is adaptable to a wide range of power and speed. In effect, the brake is a special type of centrifugal pump where the impeller and casing are specially constructed so as to waste as much energy as possible when the impeller or rotator revolves. The casing is supported on anti-friction trunnions, and is kept from rotating by the load applied at the known distance on the arms connected to the casing. The arrangement shown in the sectional view in Fig. 35 is designed for testing at high speeds of rotation, and is largely used for testing motor-car and aeroplane engines, steam turbines, etc., by builders of these engines. Both sides of the rotator contain semicircular pockets inclined forwards towards the direction of rotation, as represented by the developed sectional view in Fig. 36. Similar pockets are fixed inside the casing opposing those in the rotator. Water is supplied to the casing through a flexible hose from a water main or overhead tank, and finds its way into the pockets through the holes in the walls, as represented in Fig. 36, and as shown dotted in Fig. 35. The water is discharged at the outer circumference of the pockets into the central belt in the casing and is led away to waste. In action the water in the pockets is set into rapid rotation or vortex motion, and thus a large interchange of momentum takes place between the rotator and the casing. The moment of the forces on the casing and that on the rotator is equal, so that by measuring the moment on the casing that on the shaft is known. The work done on the water is converted into heat and the flow of water is regulated to give an outlet temperature of about 140° F. Between the faces of the rotator and the casing thin plates of metal or sluices are interposed, capable of sliding towards or away from the shaft. By rotating an external hand-wheel these sluices can be adjusted to any position. When placed fully in, all the pockets are masked by the sluice plates and in this condition only a light load is absorbed. When fully out, all the pockets are allowed to be in action, and this condition gives the maximum load at any given speed of rotation.

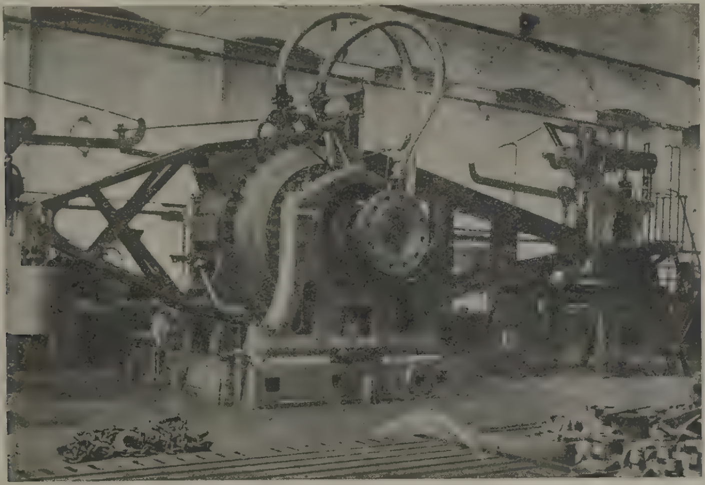
FIG. 37. FROUDE HYDRAULIC DYNAMOMETER TO ABSORB 9,000 B.H.P.

In the slow speed type of brake the power is varied by altering the quantity of water in the

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[1] *Hydraulics*, by Professor Unwin.

buckets by regulation of an outlet valve. Fig. 37 gives an outside view of this brake capable of absorbing 9,000 b.h.p. at 93 to 150 rev. per min., and is the type commonly used by builders of large gas and oil engines. The load on the arm of the brake is measured by a weighing machine of the steel-yard type, as shown in Fig. 37.

The brake horse-power is calculated as in the Prony type of brake given on page 1225.

The brakes so far discussed are classed under the general term of "absorption dynamometers," because they absorb the work done.

Transmission Dynamometers. What are termed "transmission dynamometers" not only measure the moment or the power, but also transmit the work done. The ordinary dynamo or alternator may be classed under this head, as, except for the losses in the dynamo, the power is transmitted. If the dynamo efficiency is known at different loads from previous tests, it is a comparatively simple calculation to derive the brake horse-power of the driving engine from the measured electrical energy delivered.

If the field system of a dynamo, instead of being fixed to the bed-plate, is supported concentric with the armature, the moment can be measured in the usual way, and the power measured directly and independently of the electrical output. Arrangements of this type up to powers of 250 b.h.p. at 1,000 rev. per min. are built by The Macfarlane Engineering Co., Ltd.

Torsion meters may also be included in the class of "transmission dynamometers," and are used largely on board turbine-driven ships to measure the moment transmitted through the shaft to the propeller. When a moment or torque is being transmitted through a shaft a corresponding relative twist occurs proportional to the moment transmitted. This elastic property is made use of in torsion meters. A length of shaft is selected, usually about 4 or 5 ft., and the relative twist is measured, but this relative twist is so small that some form of magnification is usually necessary to give readings easily observable. It is principally in the method of measurement or magnification adopted that the main differences occur between different arrangements of torsion meters. Many methods of magnification or measurement have been proposed and used, but the three methods now in practical use are confined to: (1) mechanical means of magnification, (2) flashlight arrangements, and (3) electrical measurements. Probably the most important example of the first type is the Denny-Edgecombe direct vision torsion meter, made by Messrs. Kelvin Bottomley & Baird, Ltd. In the second class is the Hopkinson-Thring flashlight torsion meter, and in the third class is the Ford torsion meter, both built by Messrs. Siemens Brothers & Co., Ltd. In the third class also is the Moullin torsion meter.

MATHEMATICS FOR ENGINEERS

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LESSON XXV

ADDING-UP PROBLEMS

Area under a Curve. In an earlier lesson (Lesson XI) we saw that there were many instances when the area under a curve had a meaning, and saw how to calculate it approximately by drawing the curve. It must be evident that if we have a formula for the curve, there should be a formula for the area under any portion of it. We are now able to show how the area can be calculated from the formula.

Suppose we wish to find the area under the portion AB of a curve (Fig. 98), between the ordinates AC and BD, and above the x axis. We should divide it into strips such as $PMNQ$. The area of one of these, being a small portion of the total area, A, would be denoted by δA. This area is also obtained by multiplying the height (y) by the width of the strip (δx), that is

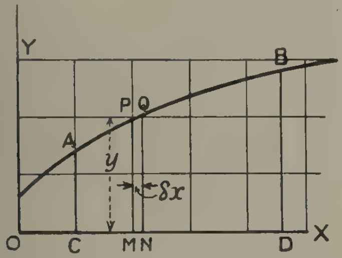

Fig. 98

$$\delta A \doteqdot y \,.\, \delta x \quad . \quad . \quad (1)$$

To find the total area A, we must add up the areas of all the strips, and using the sign Σ (sigma, the Greek S) to denote "the sum of," we may write this

$$A = \Sigma y \,.\, \delta x \quad . \quad . \quad (2)$$

(This is only saying what we are to do, but not *how* to do it.) The equation (1) is, of course, approximate since the value of y in it is neither the value at the beginning nor the value at the end, but an average value. However, if we can reduce the width of the strips indefinitely (increasing their number at the same time, of course) we shall in the limit obtain an *accurate* result. Supposing this done (and we shall see how soon) we should rewrite (2) in the form

$$A = \int y dx \quad . \quad . \quad . \quad (3)$$

where we have replaced δx by dx and Σ by $\int$ (Greek letters by English, for $\int$ is only a long S, still denoting "sum"). We have still one addition to make to (3), that is to put in the values of x between which the area we are finding lies. In the figure we are going from C to D, and if the values of x at C and D are a and b, the complete formula for the area is

$$A = \int_a^b y dx \quad . \quad . \quad (4)$$

the value of x at the start being put at the bottom, and the value at the end being put at the top. These values are called the "limits" (lower and upper limit respectively); and the complete formula is called an "integral," from "integer" meaning "whole."

So far we have done nothing more than develop a notation for "adding-up" problems, of which the area under a curve is an instance. We have now to show how to carry out the instructions, and "evaluate the integral." To do this we must return once more to the beginning, to equation (1). Dividing by δx this may be rewritten

$$\frac{\delta A}{\delta x} \doteqdot y$$

or, in the limit, when δA and δx become smaller and smaller

$$\frac{dA}{dx} = y \quad . \quad . \quad . \quad (5)$$

We now see that if we know the value of y in terms of x, i.e. the equation to the curve, we shall be able to write down the formula for A by "inverse differentiation." To see exactly how this is done, and how the initial and final values of x are to be used, we must take a particular case. We will obtain the area under the curve $y = 10x - x^2$, between the ordinates at $x = 2$ and $x = 9$. (See Fig. 99.) By (5), then, we must have

$$\frac{dA}{dx} = 10x - x^2$$

whence

$$A = 5x^2 - \tfrac{1}{3}x^3 + C \quad . \quad . \quad (6)$$

Now (5) also means that we must consider the area as growing and it starts in this case when $x = 2$. We can thus find the value of C from the fact that $A = 0$ when $x = 2$. Putting in these values,

$$0 = 5 \times 2^2 - \tfrac{1}{3} \times 2^3 + C$$

or

$$C = -(5 \times 2^2 - \tfrac{1}{3} \times 2^3)$$

where, for the present, we leave the arithmetic undone, so as not to obscure the way in which the "limits" enter into the final result. Using this value of C, we have

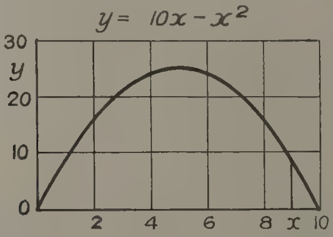

FIG. 99

$$A = 5x^2 - \tfrac{1}{3}x^3 - (5 \times 2^2 - \tfrac{1}{3} \times 2^3).$$

The meaning of this should be clear from a consideration of what has been done. It gives the area under the curve $y = 10x - x^2$, starting at $x = 2$, and finishing at x. As we want to finish when $x = 9$, we put $x = 9$ in this, and find

$$A = (5 \times 9^2 - \tfrac{1}{3} \times 9^3) - (5 \times 2^2 - \tfrac{1}{3} \times 2^3) \quad . \quad (7)$$

which, when evaluated, gives $144\tfrac{2}{3}$ as the area sought. (It can be seen from the figure that the average height of the curve between 2 and 9 is about 20, which affords a rough check upon the result.)

The student should now review the above with the object of picking out the essential steps, and formulating a rule for evaluating integrals. The final result (7), is the difference between two quantities (there bracketed), one of which contains the value 9—the upper limit, and the other contains the value 2—the lower limit. Moreover, each of them is obtained by substituting the appropriate value of x in the expression $(5x^2 - \tfrac{1}{3}x^3)$, which first made its appearance in (6), (along with a constant), and was obtained by "inverse differentiation" of the formula for y. It will thus be seen that the rule for evaluating the integral

$$\int_a^b y\,dx$$

is (provided we know the value of y in terms of x).

(*a*) **Write down the formula which, when differentiated, gives y.**

(*b*) **Subtract the value of this at the lower limit from the value at the upper limit.**

There are thus two steps—finding the new formula, and substituting values of x. (We may note that it is *not* necessary to put a constant in the new formula—and that if we do so, it will disappear upon subtracting.) It is usual to write the result of (*a*) in *square* brackets, with the limits still in evidence, so that the process of evaluation would appear thus

$$\begin{aligned} A &= \int_2^9 (10x - x^2)\,dx \\ &= \Big[5x^2 - \tfrac{1}{3}x^3\Big]_2^9 \\ &= (5 \times 9^2 - \tfrac{1}{3} \times 9^3) - (5 \times 2^2 - \tfrac{1}{3} \times 2^3) \\ &= 144\tfrac{2}{3}. \end{aligned}$$

In the same way, if we required the area under the complete arch of the curve, i.e. from $x = 0$ to $x = 10$, we should have

$$\begin{aligned} A &= \int_0^{10} (10x - x^2)\,dx \\ &= \Big[5x^2 - \tfrac{1}{3}x^3\Big]_0^{10} \\ &= (5 \times 10^2 - \tfrac{1}{3} \times 10^3) - (5 \times 0^2 - \tfrac{1}{3} \times 0^3) \\ &= 166\tfrac{2}{3}. \end{aligned}$$

If, in the same way, we required the area under $y = x^2 - 6x + 8$, from $x = 2$ to $x = 4$, we should have

$$\begin{aligned} A &= \int_2^4 (x^2 - 6x + 8)\,dx \\ &= \Big[(\tfrac{1}{3}x^3 - 3x^2 + 8x)\Big]_2^4 \\ &= (\tfrac{1}{3} \times 4^3 - 3 \times 4^2 + 8 \times 4) - (\tfrac{1}{3} \times 2^3 \\ &\qquad - 3 \times 2^2 + 8 \times 2) \\ &= \frac{16}{3} - \frac{20}{3} = -1\tfrac{1}{3}. \end{aligned}$$

We at once wonder why the result is negative, but if we draw the graph, we find that y is negative for values of x between 2 and 4, so that the curve lies *below* the x axis. (See Fig. 100.) If we evaluate the integral between the limits 0 and 4, we shall obtain the algebraic sum of the *positive* area OAB and the *negative* area BCD in the figure.

$$\int_0^4 (x^2 - 6x + 8)\,dx$$

$$= \left[\tfrac{1}{3}x^3 - 3x^2 + 8x\right]_0^4$$

$$= \left(\frac{16}{3}\right) - (0) = \frac{16}{3}$$

$$\int_0^2 (x^2 - 6x + 8)\,dx$$

$$= \left[\tfrac{1}{3}x^3 - 3x^2 + 8x\right]_0^2$$

$$= \left(\frac{20}{3}\right) - (0) = \frac{20}{3} = \text{area } OAB.$$

Another case in which the integral may come out negative, is seen by remembering that an integral is a sum (more precisely, the *limit* of a sum), each term of which is the product of y and dx. Each term, and therefore the sum, will be negative if y is positive, and dx is negative, i.e. if we go *backwards* in the x direction. As an instance,

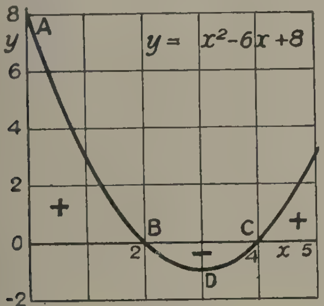

FIG. 100

$$\int_2^0 (x^2 - 6x + 8)\,dx = \left[\tfrac{1}{3}x^3 - 3x^2 + 8x\right]_2^3$$

$$= (0) - \left(\frac{20}{3}\right) = -\frac{20}{3}$$

giving a negative result, as expected. It will be seen, as a result of the argument above, that inverting the limits changes the sign (but not the numerical value) of any integral. In symbols,

$$\int_a^b y\,.\,dx = -\int_b^a y\,.\,dx$$

We also see, by the same argument, that if y and dx are *both* negative, the integral will be positive. Thus the student may verify that

$$\int_4^2 (x^2 - 6x + 8)\,dx = +1\tfrac{1}{3}.$$

It will be found that these rules for the sign of an area may be summarized by saying that, if we go along the x axis in the direction of integration (starting at the lower, and proceeding towards the upper, limit) an area on the *left* is *positive*, while an area on the *right* is *negative*.

We give a few examples in which the "integrand" (i.e. the expression to be integrated) needs treatment before the integration can be effected by rule.

$$\int_1^9 5\sqrt{x}\,dx = \int_1^9 5x^{\frac{1}{2}}\,dx$$

$$= \left[\frac{5}{1\frac{1}{2}}\,x^{1\frac{1}{2}}\right]_1^9 = \left[\frac{10}{3}\,x\sqrt{x}\right]_1^9$$

$$= \left(\frac{10}{3} \times 9 \times 3\right) - \left(\frac{10}{3} \times 1 \times 1\right)$$

$$= \frac{260}{3} = 86\tfrac{2}{3}$$

$$\int_2^4 \frac{6}{x^2}\,dx = \int_2^4 6x^{-2}\,dx$$

$$= \left[\frac{6}{-1}\,x^{-1}\right]_2^4 = \left[-\frac{6}{x}\right]_2^4$$

$$= \left(-\frac{6}{4}\right) - \left(-\frac{6}{2}\right)$$

$$= -1\tfrac{1}{2} + 3 = 1\tfrac{1}{2}.$$

It should be noticed in these, that we have expressed the integrand in terms of indices before applying the rule, but have expressed our results in the simplest form for calculation before we have substituted the values at the limits. The student should verify by drawing the graphs that these two integrals do give the areas under the curves $y = 5\sqrt{x}$ and $y = \dfrac{6}{x^2}$ respectively, between the appropriate limits.

FURTHER EXAMPLES. 1. In the stretching of a bar in tension, a certain amount of work is done. We know that this is represented by the area under a force-extension graph, i.e. if F is the force (tension) when the extension is x, and e is the final extension, the work done is

$$W = \int_0^e F\,.\,dx$$

But, E being Young's Modulus, l the length, and A the cross-sectional area, we have

$$F = \frac{A\,.\,E\,.\,x}{l}$$

Therefore

$$W = \int_0^e \frac{A \,.\, E}{l}\, x \,.\, dx = \left[\frac{A \,.\, E}{l}\,\frac{x^2}{2}\right]_0^e$$
$$= \frac{A \,.\, E \,.\, e^2}{2l}$$

Also, if P denotes the final tension, $P = \dfrac{A \,.\, E \,.\, e}{l}$

and so $W = \frac{1}{2}Pe$.

It can be shown, in exactly the same way, that whenever the displacement is proportional to the force producing it (and that is, whenever Hooke's Law is obeyed) the work is given by half the product of the final force and the final displacement.

2. If $pv^{1.4} = 240$, find the value of $\int_{0.4}^{2.2} p \,.\, dv$.

(If p is pressure, and v volume, of a quantity of gas, then $\int_{v_1}^{v_2} p \,.\, dv$ gives the work done in expansion from v_1 to v_2, as is shown in the articles on Heat Engines.)

Here, since $pv^{1.4} = 240$, $p = \dfrac{240}{v^{1.4}} = 240v^{-1.4}$

$$\text{Thus } \int_{0.4}^{2.2} p \,.\, dv = \int_{0.4}^{2.2} 240v^{-1.4}dv$$
$$= \left[\frac{240}{-0.4}\, v^{-0.4}\right]_{0.4}^{2.2}$$
$$= \left[-\frac{600}{v^{0.4}}\right]_{0.4}^{2.2}$$
$$= \left(-\frac{600}{2.2^{0.4}}\right) - \left(-\frac{600}{0.4^{0.4}}\right)$$
$$= -437.7 + 865.8 = 428.1.$$

EXERCISE No. 39

1. Evaluate the following definite integrals—

(a) $\int_0^4 3x^2dx$.

(b) $\int_1^5 (2x + 1)\, dx$.

(c) $\int_1^3 (x^2 + x + 1)\, dx$.

(d) $\int_{-1}^1 (3 - x^2)\, dx$.

(e) $\int_4^2 (x - 3)\, dx$.

(f) $\int_{0.5}^2 \frac{8}{x^3}\, dx$

(g) $\int_9^{16} \frac{10}{\sqrt{x}}\, dx$.

(h) $\int_{1.2}^{3.5} \frac{300}{x^{1.5}}\, dx$.

2. Draw the graph of $y = 4 + \frac{1}{2}x$, from $x = 0$ to $x = 6$. Find the area underneath it between these limits, (a) by integration, and (b) by mensuration formulae.

3. Show that the area under the curve $y = ax^2$, from $x = 0$ to $y = b$ is $\frac{1}{3}bh$, h being the value of y when $x = b$.

4. Show similarly, for the curve $y = ax^n$ (where n is positive) that the area is $\dfrac{1}{n+1}\, bh$. (This result is known as Wallis' Law, and was discovered by Wallis before the invention of the calculus.)

5. Find the area between the curve $y = 2x - \dfrac{x^2}{20}$ and the line $y = 25$, from $x = 0$ to $x = 40$. Considering the curve as an arch, and the line as the road above, x and y being in feet, find the number of cubic yards of masonry in the bridge if the width is 30 ft.

6. If $pv^{1.3} = 2060$, evaluate $\int_{1.1}^{4.9} p \,.\, dv$.

7. If $y = \sqrt{100 - x^2}$, find the value of $\int_{-10}^{10} \pi y^2\, dx$. (It will be seen later that this integral represents the volume of a sphere of radius 10.)

ANSWERS TO EXERCISES NOS. 37 AND 38

(37)

1. 4·5 lb.
2. 9·167. . ., 6·714. . . , 3·917. . . .
3. 2·1316. . . .

(38)

1. (a) $x^4 + C$; (b) $1\frac{1}{2}x^2 + 2x + C$;
   (c) $\frac{1}{5}x^5 - \frac{3}{4}x^4 + \frac{1}{2}x^2 + C$;
   (d) $0.8x^{2.5} - 10x^{0.4} + C$;
   (e) $-\dfrac{1}{2x^2} + C$; (f) $\dfrac{10}{3}x^{1.5} + 4x^{0.5} + C$.
2. $y = \frac{1}{8}x^4 - x^2 + 4$; 4, 20.
3. $v = 4.4\left(t - \dfrac{t^2}{60}\right)$; $s = 4.4\left(\dfrac{t^2}{2} - \dfrac{t^3}{180}\right)$; 49·5 ft./sec., 412·5 ft.
4. $y = 4 + 0.01x^2$; 50 and 64 tons (each chain); 38°40′.
5. $t = 360 - 120\sqrt{h}$; 360 sec. = 6 min.

POWER TRANSMISSION BY BELTING

By W. G. Dunkley, B.Sc.

LESSON X

POSITION OF DRIVING SIDE, VERTICAL DRIVES, SHORT-CENTRE DRIVES, LENIX DRIVE

Position of Driving Slide. It is highly important that, wherever possible, the underside of the belt should be the driving side. It will be readily seen that, if the slack side is underneath, the weight of the belt will cause the belt to hang away from the pulley and so reduce the angle of lap. On the other hand, if the slack side is on top, the slack of the belt increases the arc of contact. This applies to horizontal and inclined drives. With an inclined drive, if the slack side is underneath, the sag of the slack side will reduce the arc of contact on the lower pulley more than on the upper pulley, and if the lower pulley is the smaller pulley the reduction in the arc of contact may be serious. As the belt stretches there will be a corresponding reduction in the arc of contact, as well as loss of tension. A consideration of the above points will show that the position of the driving side is of real importance.

Vertical Drives. Vertical drives and drives approaching the vertical should be avoided if possible. It will be seen that the weight of the belt is carried entirely by the upper pulley, and the belt is tending to hang away from the surface of the lower pulley. As soon, therefore, as the belt tension falls, owing to the permanent stretching of the belt, the grip on the lower pulley, particularly if this is the smaller pulley, will soon be reduced and slip will commence.

Short-centre Drives. With short-centre drives the small angle of lap on the smaller pulley is a point of disadvantage which is readily obvious. We have seen that it is the gradual permanent stretching of the belt which really kills the belt in time. Consider the varying conditions which a given part of the belt is continually enduring. On entering the tight side it is stretched, as it reaches the pulley it is bent round the pulley and its inner fibres are compressed and its outer fibres stretched; as it leaves the pulley it is bent straight again, which changes the stress conditions and its tension is reduced, which means that it has contracted. It reaches the other pulley and is bent again, straightened again as it leaves the pulley and again stretched, and so the cycle commences once more.

It is clear, therefore, that a short belt is being bent, straightened, stretched, and contracted with greater frequency than a long belt running at the same speed. We should not, therefore, expect the same length of life out of the shorter belt.

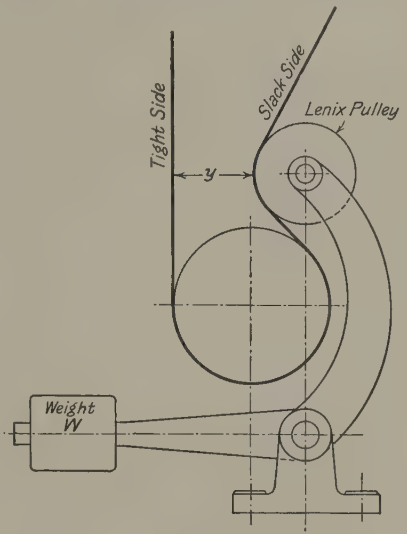

Fig. 8

Lenix Drive

Description. In general, as we have noted, vertical drives and short-centre drives should be avoided. With the Lenix drive it is quite permissible to use vertical drives and short-centre drives. This type of drive is used very frequently where a self-contained drive is desired, where the centre distance would be too short for an ordinary drive. The principle of the drive is the use of a jockey pulley riding on the slack side of the belt, by means of which pulley the necessary initial tension in the belt is produced and maintained. The principle is illustrated in Fig. 8. The means of applying the pulley may, of course, be varied to suit the circumstances. The use of the Lenix pulley should not be confused with the use of a tension pulley acting on the tight side of the belt, and used to increase the belt tension as the belt stretches and slip starts.

Application of the Lenix Pulley. The Lenix pulley should be applied as near as possible to the smaller pulley, so as to increase the angle of lap as much as possible on this pulley. The length of the belt should be such that the distance y, Fig. 8, is about equal to the radius of the smaller pulley.

ORDINARY AND LENIX DRIVE COMPARED

EXAMPLE 19. We desire to transmit 20 h.p. from a 10 in. diameter motor pulley, running at 900 r.p.m. Driven pulley, 50 in. diameter. Centre distance, 5 ft. Determine the belts for an ordinary drive and for a Lenix drive. Compare the belt tensions in each case.

First we require to set down the two drives to scale, and to measure the angle of lap in each case. The smaller angle of lap in each case should be taken.

Ordinary drive. Angle of lap, 135°
Lenix drive. Angle of lap, 235°

1. *Ordinary drive*—

$$\text{Log}\ \frac{T}{t} = 0{\cdot}007578 \times 0{\cdot}4 \times 135$$

$$\text{Then}\ \frac{T}{t} = 2{\cdot}57.$$

Lenix drive—

$$\text{Log}\ \frac{T}{t} = 0{\cdot}007578 \times 0{\cdot}4 \times 235$$

$$\text{Then}\ \frac{T}{t} = 5{\cdot}16.$$

2. Effective driving pull

$$P = \frac{H \times 126{,}000}{D \times N}$$

$$= \frac{20 \times 126{,}000}{10 \times 900} = 280 \text{ lb.}$$

This is the same for both drives.

$$3.\quad T = \frac{\dfrac{T}{t} \times P}{\left(\dfrac{T}{t} - 1\right)}$$

$$\textit{Ordinary drive}\quad T = \frac{2{\cdot}57 \times 280}{1{\cdot}57}$$

$$= 458 \text{ lb.}$$

$$\textit{Lenix drive}\quad T = \frac{5{\cdot}16 \times 280}{4{\cdot}16}$$

$$= 347 \text{ lb.}$$

4. *Ordinary drive* $t = 458 - 280 = 178$ lb.
Lenix drive $t = 347 - 290 = 67$ lb.

5. We shall neglect the effect of centrifugal force. Taking a single cemented belt we allow 400 lb./sq. in.

$$\textit{Ordinary drive}\quad x \times \tfrac{3}{16} \times 400 = T = 458$$

$$\text{Width of belt}\ x = \frac{458 \times 16}{400 \times 3} = 6 \text{ in.}$$

$$\textit{Lenix drive}\quad x \times \tfrac{3}{16} \times 400 = T = 347$$

$$x = \frac{347 \times 16}{400 \times 3} = 4\tfrac{1}{2} \text{ in.}$$

The above figures do not show all the advantages given by the Lenix drive. We took the same coefficient of friction 0·4 in each case. The belt is working under such better conditions in the Lenix drive as regards lap on the pulley, that we could reasonably allow a higher value of μ which would allow us to adopt a still smaller belt in comparison. This point will be better appreciated by considering the following comparison as regards slip.

Slipping Conditions Compared. The following remarks bring out a very important feature of the Lenix drive, and the point should be considered carefully. In the ordinary drive, we must have a minimum initial belt tension if the belt is not to slip, and, as we have seen, the permanent stretch of the belt gradually reduces the initial tension until slipping commences, and it becomes necessary to shorten the belt to obtain the necessary initial tension.

Now notice that, with the Lenix pulley, the initial tension is produced by the weight W. Thus, as the belt stretches it does not lose its tension, because the weight W is still maintaining the tension; the stretch of the belt just allows the pulley to move to a lower position. We see, therefore, that we do not expect slip except that caused by overload.

Limiting Centre Distance. With an ordinary drive, the centre distance is limited by the necessity of having a reasonable angle of lap on the smaller pulley. With the Lenix drive, the centre distance is only limited by the space required between the pulleys for the Lenix pulley.

(*Continued on page* 1240)

APPLIED MECHANICS

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LESSON XXII

Friction

Let us consider two bodies, A and B, Fig. 131, which are in contact at a point C. Whilst there is no tendency for motion to take place between the two bodies, the *contact force* is normal to the common tangent plane passing through C. When, however, one body moves, or tends to move relatively to the other (contact being maintained), Fig. 132, the contact force acts along a line inclined to the tangent plane as shown. It is here that we become aware of the *force of friction*, which is the component of the contact force in a direction parallel to the tangent plane, that is,

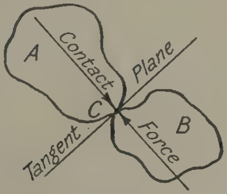


Fig. 131

Force of friction

$$= \text{contact force} \times \sin\theta \qquad . \quad . \quad . \quad (1)$$

It is invariably found that *friction opposes motion*, and hence the direction in which motion occurs or tends to occur will determine the side of the normal on which the contact force will lie.

Coefficient of Friction. When the force P tending to make one body move over another has reached such a value that motion commences or is about to commence, then the ratio

$$\frac{\text{force required to produce motion}}{\text{normal component of contact force}} = \text{coefficient of friction},$$

or, as in Fig. 132, coefficient of friction $= \tan\theta = \mu$ (usual symbol).

The *laws of friction* seek to state the relations between the force of friction and, (1) the area of the surfaces in contact, (2) the normal pressure between the surfaces in contact, (3) the speed of rubbing, and (4) the nature of the surfaces in contact. The relations are only of a simple nature when we do not exceed certain limits, and, at all times, we must exercise caution in the application of these laws.

Law 1. The force of friction is independent of the area of the surfaces in contact.

Law 2. The force of friction is directly proportional to the normal pressure between the surfaces.

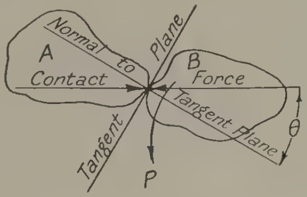


Fig. 132

Law 3. The force of friction is independent of the speed of rubbing.

Law 4. The force of friction depends upon the nature of the surfaces in contact.

Inclined Plane

Case 1. The force P acts parallel to the horizontal base causing motion *up* the plane. Since motion is *up* the plane, the contact force will act along FD, Fig. 133, which makes an angle $(\theta + \alpha)$ with the vertical.

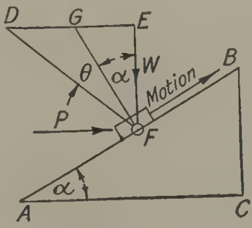


Fig. 133

From the triangle DEF, we have—

$$\frac{DE}{EF} = \frac{P}{W} = \tan(\theta + \alpha)$$

or, $$P = W\tan(\theta + \alpha) \qquad . \quad . \quad . \quad (1)$$

If friction were absent we should have—

$$\frac{GE}{EF} = \frac{P}{W} = \tan \alpha$$

or,

$$P = W \tan \alpha \qquad . \qquad . \quad (2)$$

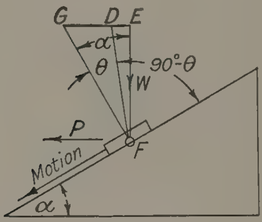


FIG. 134

hence, as a lifting device (square thread screw), the efficiency of the inclined plane

$$= \frac{(2)}{(1)} = \frac{\tan \alpha}{\tan (\theta + \alpha)} \qquad . \qquad . \quad (3)$$

Case 2. The force P acts parallel to the horizontal base causing motion *down* the plane. In this case, the contact force FD (Fig. 134) is inclined at $(\alpha - \theta)$ to the vertical. From the triangle DEF we have—

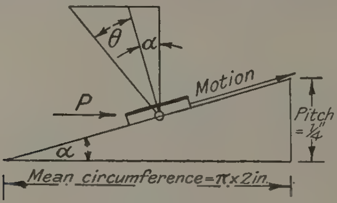


FIG. 135
(Not to scale)

$$\frac{DE}{EF} = \frac{P}{W} = \tan (\alpha - \theta)$$

$$\therefore \; P = W \tan (\alpha - \theta) \qquad . \qquad . \quad (4)$$

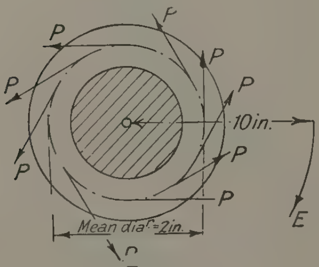


FIG. 136
(Not to scale)

EXAMPLE. The mean diameter of a square thread screw is 2 in., and the pitch is $\frac{1}{4}$ in. If the coefficient of friction = 0·25, find the effort required at the end of a tommy bar 10 in. long to raise an axial load of $\frac{1}{4}$ ton. Find also the efficiency of the screw (Fig. 135).

$$\tan \alpha = \frac{\frac{1}{4}}{\pi \times 2} = 0{\cdot}03978, \; \therefore \; \alpha = 2° - 16'{\cdot}8$$

$$\tan \theta = \mu = 0{\cdot}25 \quad \therefore \; \theta = 14° - 2'{\cdot}2$$

$$P = W \tan (\alpha + \theta) = 560 \; \tan 16°{\cdot}19' \text{ or}$$

$$P = 163{\cdot}9 \text{ lb.}$$

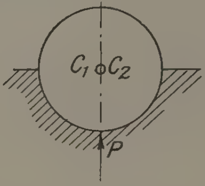


FIG. 137

Taking moments of forces about the centre of the screw, Fig. 136, we have—

$$E \times 10 \text{ in.} = P \times 1 \text{ in.}$$

$$\therefore \; E = \frac{163{\cdot}9}{10} = 16{\cdot}39 \text{ lb.}$$

$$\text{Efficiency} = \frac{\tan \alpha}{\tan (\theta + \alpha)} = \frac{0{\cdot}03978}{0{\cdot}2927}$$

$$= {\cdot}135, \text{ or } 13{\cdot}5 \text{ per cent.}$$

$$\text{or, efficiency} = \frac{\text{useful work}}{\text{total work}} = \frac{560 \times \frac{1}{4}}{16{\cdot}39 \times 2\pi \times 10}$$

$$= {\cdot}135, \text{ or } 13{\cdot}5 \text{ per cent.}$$

JOURNAL FRICTION

When a shaft is at rest in its bearing the load W on the shaft acts vertically downwards through the centre of the bearing, and we may consider the shaft load W to be taken by the bearing at the point P (Fig. 137). When rotation of the shaft takes place, the friction between the surfaces in contact causes the shaft to climb up into some such position as shown in Fig. 138.

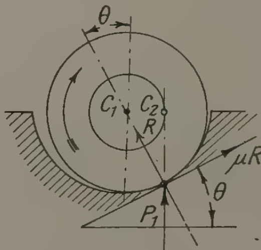


FIG. 138

The point P is now displaced to P_1, and the reaction vertically upwards is tangential to a circle of radius C_1C_2. The moment of W about $C_1 = W \times C_1C_2 = W \times r \sin\theta$ which, since θ is small, may be written $Wr \tan\theta$ or friction moment $= Wr\mu$.

EXAMPLE. A shaft, 8 in. diameter and running at 180 r.p.m., carries a fly-wheel whose weight is 6 tons. The radius of gyration of the wheel is 5½ ft. If μ for the shaft in its bearings is 0·075, determine how many revolutions the shaft and wheel will make before coming to rest, and the time of coming to rest after the driving torque has ceased to act, the shaft being subjected to the frictional retarding torque only which may be assumed constant. Neglect the kinetic energy of the shaft.

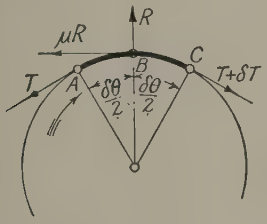


FIG. 139

Kinetic energy stored in wheel at 180 r.p.m.

$$= \tfrac{1}{2} I\omega^2 = \tfrac{1}{2}\frac{W}{g} k \times \omega^2 \quad . \quad . \quad . \quad . \quad (1)$$

Where W = weight of wheel; k = radius of gyration of wheel and ω = radians per sec.

$$= \frac{180 \times 2\pi}{60} = 6\pi$$

Work expended in friction per rev.

$$= Wr\mu \times 2\pi \quad . \quad . \quad . \quad . \quad . \quad (2)$$

∴ revolutions made by shaft before stopping

$$= \frac{(1)}{(2)} = \frac{Wk^2\omega^2}{2g \times Wr\mu 2\pi}$$

$$= \frac{5{\cdot}5 \times 5{\cdot}5 \times 36\pi^2 \times 12 \times 1000}{2 \times 32{\cdot}2 \times 4 \times 75 \times 2\pi} = 1047 \text{ revs.}$$

Since the initial speed is 180 r.p.m. and the final speed is zero, therefore the average speed

= 90 r.p.m. = 1·5 r.p.s.

∴ time required to bring wheel to rest

$$= \frac{1047}{1{\cdot}5} = 698 \text{ sec.} = 11 \text{ min. } 38 \text{ sec.}$$

COIL FRICTION

When a flexible strap is wrapped around a cylinder there is found to be a difference between the tensions applied to the two ends at the instant when slipping of the strap on the cylinder is about to take place. Let us take an element of length of strap which laps the arc ABC and subtends the angle $\delta\theta$ (θ being in radians) (Fig. 139). The forces acting on this piece of strap are the two tensions T and $T + \delta T$, the normal reaction R and the frictional resistance μR. Drawing the force polygon we obtain Fig. 140.

Resolving forces vertically we have

$$R = T \sin\frac{\delta\theta}{2} + (T + \delta T)\sin\frac{\delta\theta}{2}$$

$$R = T \sin\frac{\delta\theta}{2} + T \sin\frac{\delta\theta}{2} + \delta T . \sin\frac{\delta\theta}{2}$$

$$R = 2T \sin\frac{\delta\theta}{2} \quad \left[\delta T . \sin\frac{\delta\theta}{2} \text{ being negligible}\right]$$

[$\sin\frac{\delta\theta}{2}$ may be written $\frac{\delta\theta}{2}$ radians when $\frac{\delta\theta}{2}$ is small]

$$\text{or, } R = T . \delta\theta \quad . \quad . \quad . \quad (1)$$

Again, resolving forces horizontally, we have

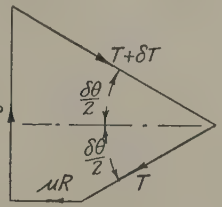


FIG. 140

$$\mu R + T \cos\frac{\delta\theta}{2} = (T + \delta T)\cos\frac{\delta\theta}{2}$$

$$\text{or} \quad \mu R = \delta T \cos\frac{\delta\theta}{2}$$

and since $\frac{\delta\theta}{2}$ is a small angle, $\cos\frac{\delta\theta}{2}$ may be taken as equal to 1

$$\therefore \mu R = \delta T \quad . \quad . \quad . \quad (2)$$

By the substitution of (2) in (1) we obtain

$$\frac{\delta T}{T} = \mu . \delta\theta \quad . \quad . \quad . \quad (3)$$

Integrating between the appropriate limits we may write

$$\int_{T_r}^{T_d} \frac{\delta T}{T} = \mu \int_0^\theta \delta\theta$$

$$\therefore \log_e \frac{T_d}{T_r} = \mu\theta$$

$$\text{or } \frac{T_d}{T_r} = e^{\mu\theta} \quad . \quad . \quad . \quad (4)$$

Example. A rope is coiled three times around a bollard. Taking μ as 0·3, determine the least force with which one end of the rope must be pulled when a load of 3 tons is attached to the other end and slipping is just prevented.

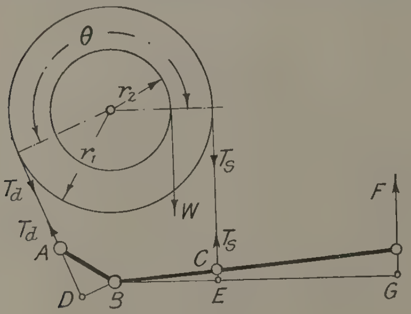


Fig. 141

$$\frac{T_d}{T_s} = e^{\mu\theta}$$

$\therefore$ $\log. T_d - \log. T_s = \mu\theta \log. e$

or $\log. T_s = \log. T_d - \mu\theta \log. e$

or $\log. T_s = 1{\cdot}3712$

or $T_s = 23{\cdot}51$ lb.

where $T_d = (3 \times 2240)$ lb.

$\mu = 0{\cdot}3$

$\theta = (3 \times 2\pi)$ radians

$T_s =$ least pull required

Band Brakes. A band brake is one having a flexible strap, which is wrapped partly or wholly around the circumference of a brake drum. Two arrangements are commonly used : (1) One end of the strap is pinned to the lever at A, Fig. 141, and the other end to another point C in the lever ; (2) One end of the strap is pinned to the lever as at A, and the other end is pinned to some fixed point on the machine frame, which point may be the axis pin of the lever itself. With T_d and T_s as the respective pulls in the ends of the strap, we have

$$\text{Friction torque} = (T_d - T_s)r_1$$
$$= W \times r_2 \quad . \quad . \quad . \quad (1)$$

A second expression is obtained by taking moments of forces about the fulcrum pin B ; thus

$$(F \times BG) + (T_s \times BE)$$
$$= (T_d \times BD) \quad . \quad . \quad . \quad . \quad (2)$$

Also, we may use the expression

$$\frac{T_d}{T_s} = e^{\mu\theta} \quad . \quad . \quad . \quad (3)$$

Example. A band brake is of the pattern shown in Fig. 141, and the dimensions are as follows: $BE = 3$ in., $BG = 24$ in., $BD = 2{\cdot}5$ in., $r_1 = 10$ in., $r_2 = 8$ in., $\theta = 220°$. If $\mu = 0{\cdot}36$, determine the vertical force which must be applied at F to just hold up the load of $W = 3$ tons.

Expressions (1), (2), (3) above are available ; by substitution in (1) we have

$$T_d - T_s = \frac{3 \times 8}{10} = 2{\cdot}4 \text{ tons} \quad . \quad . \quad (4)$$

$$\text{Also} \quad \frac{T_d}{T_s} = e^{\mu\theta}$$

$$\therefore \log \frac{T_d}{T_s} = \mu\theta \log e$$

$$= \frac{0{\cdot}36 \times 220 \times 2\pi \times {\cdot}4343}{360}$$

$$= 0{\cdot}6004$$

$$\text{or} \quad \frac{T_d}{T_s} = 3{\cdot}985 \quad . \quad . \quad . \quad . \quad (5)$$

Combining (4) and (5) we have

$$T_s = \frac{2{\cdot}4}{2{\cdot}985} \text{ tons}$$

$$\text{and} \quad T_d = \frac{3{\cdot}985 \times 2{\cdot}4}{2{\cdot}985}$$

By substitution in (2) we obtain

$$F \times 24 = \frac{3{\cdot}985 \times 2{\cdot}4 \times 2{\cdot}5}{2{\cdot}985} - \frac{2{\cdot}4 \times 3}{2{\cdot}985}$$

$$\text{or} \quad F = 0{\cdot}2332 \text{ tons.}$$

FRICTION DRIVES

Plate Clutches. This form of friction drive is of increasing importance in automobile transmission gears. In the usual arrangement, a thin metal (steel) disc or discs are placed between annular rings of some friction fabric suitably supported so that when pressure is applied between the surfaces, the rotating driving member "picks up" the other and so transmits its motion. Let Fig. 142 represent the surface of an annular ring which is in contact with a similar annular ring. When unit pressure p is applied between the surfaces, the load on an elementary ring of radial width δx, as in Fig. 142, is

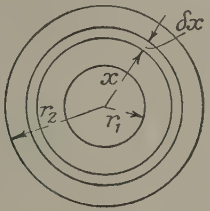


FIG. 142

$$p \times 2\pi x \,.\, \delta x \qquad . \quad . \quad (1)$$

Let us suppose that *wear* is to take place *uniformly distributed* all over the friction surfaces and, we may assume that wear is proportional to (1), v the speed of rotation of one surface over the other, and (2) p the unit pressure between the surfaces in contact; hence we may write

$$p \times v \quad = \text{constant}$$

or, since $v \propto x \; \therefore p \times x = \text{constant} = k$ (say)

or $$p = \frac{k}{x} \qquad . \quad . \quad (2)$$

By substitution of (2) in (1) and integrating we obtain

$$\text{Total clamping load} = P = \int_{r_1}^{r_2} \frac{k}{x} \times 2\pi x dx$$

$$\therefore \; P = k 2\pi \,(r_2 - r_1)$$

or, $$k = \frac{P}{2\pi \,(r_2 - r_1)} \qquad . \quad . \quad (3)$$

Further, the friction torque of an elementary ring about the axis

$$= \mu p \times 2\pi x \,.\, \delta x \times x\,;$$

and $$p = \frac{k}{x} = \frac{P}{x \,.\, 2\pi(r_2 - r_1)}$$

Substituting and integrating we obtain

Total friction torque

$$= \int_{r_1}^{r_2} \mu \frac{P}{x} \times \frac{2\pi x \times x \,.\, dx}{2\pi(r_2 - r_1)}$$

Total friction torque

$$= \frac{\mu \, P(r_2 + r_1)}{2} \qquad . \quad . \quad . \quad (4)$$

EXAMPLE. A single plate clutch (two friction faces) is required to transmit 30 h.p. at 1800 r.p.m. If $r_1 = \frac{2}{3} r_2$, determine these radii when $\mu = 0{\cdot}4$ and the clamping load is 400 lb.

$$\frac{\text{Torque} \times 2\pi n}{12 \times 33{,}000} = \text{h.p.}$$

$\therefore$ torque per friction face

$$= \frac{30 \times 12 \times 33{,}000}{2\pi \times 1800 \times 2} \qquad . \quad . \quad (1)$$

Also torque

$$= \frac{\mu P(r_2 + r_1)}{2} \qquad . \quad . \quad . \quad (2)$$

and $r_2 = \dfrac{3}{2} r_1$.

$$\therefore \; (1) = (2), \text{ or}$$

$$\frac{30 \times 12 \times 33{,}000}{2\pi \times 1800 \times 2} = \frac{4 \times 400 \times 5 r_1}{10 \times 2 \times 2}$$

$\therefore$ $\qquad r_1 = 2{\cdot}626$ in.

and $\qquad r_2 = 3{\cdot}939$ in.

CONE CLUTCHES. Again we proceed on the assumption of *uniform wear* and, with the notation previously used, we can write

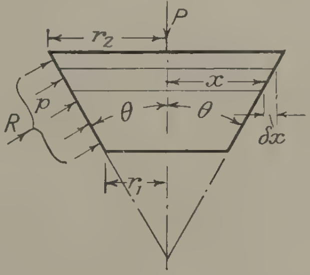


FIG. 143

$$p = \frac{k}{x} \qquad . \quad . \quad . \quad (1)$$

Considering (Fig 143) an elementary ring of radius x and width, measured along a generator of the cone, $\delta x \,.\, \operatorname{cosec} \theta$, subjected to a normal unit pressure p, the load on this ring $= p \,.\, \delta x \,.\, \operatorname{cosec}\theta \times 2\pi x$; $\therefore$ total load normal to cone surface

$$= R = \int_{r_1}^{r_2} p \,.\, 2\pi x \,.\, \operatorname{cosec}\theta \,.\, dx \quad . \quad . \quad (2)$$

and on substituting (1) in (2) and integrating we obtain

$$R = 2\pi k \operatorname{cosec}\theta\,(r_2 - r_1)$$

$$\therefore \quad k = \frac{R}{2\pi \operatorname{cosec}\theta\,(r_2 - r_1)} \quad . \quad . \quad . \quad (3)$$

The friction torque on this elementary ring about the axis of the cone

$$= \mu \,.\, p \,.\, 2\pi x \,.\, \operatorname{cosec}\theta\, \delta x \times x$$

$$= \mu \frac{k}{x} \operatorname{cosec}\theta\, 2\pi x^2 \,.\, \delta x$$

and total friction torque

$$= \int_{r_1}^{r_2} \mu k \operatorname{cosec}\theta\, 2\pi x dx$$

$$= \frac{\mu R \times 2\pi \operatorname{cosec}\theta}{2\pi \operatorname{cosec}\theta\,(r_2 - r_1)} \times \frac{(r_2^2 - r_1^2)}{2}$$

$$= \frac{\mu R\,(r_2 + r_1)}{2}; \text{ and } \frac{P}{R} = \sin\theta$$

$$= \frac{\mu P\,(r_2 + r_1)}{2 \sin\theta} \quad . \quad . \quad . \quad . \quad (4)$$

Example. The semi-angle of a cone clutch is 7°, and the smaller diameter is 20 in. Determine the larger diameter of the cone and the axial length of the clutch required to transmit 120 h.p. at 1500 r.p.m. when the clamping pressure is 350 lb. Take $\mu = 0{\cdot}17$.

$$\text{Torque} = \frac{\text{h.p.} \times 12 \times 33{,}000}{2\pi \times n}$$

$$,, \quad = \frac{120 \times 12 \times 33{,}000}{2\pi \times 1500} \text{ lb.-in.} \quad . \quad (1)$$

$$\text{Also torque} = \frac{\mu P\,(r_2 + r_1)}{2 \sin\theta}$$

$$\text{or torque} = \frac{\cdot 17 \times 350\,(r_2 + 10)}{2 \times \cdot 1219} \quad . \quad (2)$$

We may now equate (1) = (2)

$$\text{or,} \quad \frac{120 \times 12 \times 33{,}000}{2\pi \times 1500} = \frac{\cdot 17 \times 350\,(r_2 + 10)}{2 \times \cdot 1219}$$

$$\therefore \; r_2 = 10{\cdot}65 \text{ in.}$$

or larger diameter $= 21{\cdot}3$ in.

$$\text{Axial length of cone} = \frac{\cdot 65}{\tan 7^\circ} = \frac{\cdot 65}{\cdot 1228}$$

$$= 5{\cdot}21 \text{ in.}$$

Note. Plate and cone clutches may also be treated on the assumption of a uniform distribution of pressure and, as this is rather easier than the above text, it is left as an exercise for the student.

POWER TRANSMISSION BY BELTING

(*Continued from page* 1234)

Speed Ratios. With the ordinary drive the speed ratio is limited, a ratio of about 6 to 1 being the limit. With the Lenix pulley, speed ratios of 30 to 1 have been successfully used. This is, of course, an extreme case.

Stopping and Starting Devices. The Lenix pulley can be used as a stopping and starting device, since the belt becomes quite slack when the Lenix pulley is lifted from the belt.

Study. Compare the belts necessary for an ordinary and Lenix drive. Motor, 15 h.p. at 1,000 r.p.m. Pulleys, 9in. and 40 in. diameter. Centre distance, 5 ft. Take $u = 0{\cdot}4$ for ordinary drive, $\mu = 0{\cdot}6$ for the Lenix drive. Belt, cemented, single.

STRUCTURAL STEELWORK

By W. Arnold Green, M.A., B.Sc., A.M.Inst.C.E., M.I.Struct.E.

LESSON V

STRUCTURAL STEEL—(*contd.*)

Wind Load on Steel Frame Buildings. In many steel frame buildings special calculations for the stability of the structure as a whole for resisting wind are not necessary, as the floors act as deep horizontal girders and transfer wind loads to cross walls, which prevent any racking. If stiffening walls are absent and it is not feasible to employ diagonal members to form, with the beams and columns, vertical braced cantilevers, the side load will produce bending moments in each column which must be taken up at each floor by the beams connecting them.

On the assumption of absolutely rigid connections and knowing the relative stiffnesses of beams and columns, the stresses in the structure can be completely determined, but the process is complicated and hardly justified by the nature of the assumptions made.

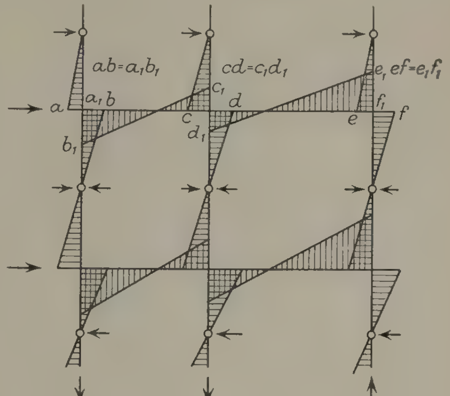
A sufficiently accurate method of analysis for calculating the stresses in the individual columns due to shear on the structure regarded as a cantilever fixed at the ground-level, is to assume that the wind loads are applied at floor levels, and that each column is pin connected at its mid-height, as indicated in Fig. 25, which shows the centre lines of beams and columns of one bay in one storey of a tall building carried by three lines of columns.

As the movement of each pin in a horizontal line may be assumed to be the same and proportional to the applied load divided by the moment of inertia of the column, the total shear at the line of pins is assumed to be divided between the columns in proportion to their moments of inertia, so that the greatest horizontal load comes on to the stiffest column.

The resulting bending moments will be as indicated in the figure, the diagrams being drawn on the tension sides of the members.

At any internal joint, the sum of the column bending moments is assumed divided between the beams in the proportion of their moments of inertia divided by their lengths. The end bending moment in the beam joined to the outside column is the sum of the bending moments in the column.

The wind load will also cause tension in the columns on the windward side and compression on the leeward side. The outside columns will naturally receive the maximum stresses. These vertical tensile and compressive stresses, assumed constant throughout the section of each column, may be found as follows: (1) find the centroid of the column areas, (2) calculate the moment of inertia of the areas about the centroid, (3) divide the bending moment at the horizontal section considered (which should be in a line of "pins") by this moment of inertia, (4) multiply the quantity thus found by the distance from the centroid to find the stress in the column (tensile or compressive), (5) multiply the stress so found by the column area.



Wind Bending Moment Diagram.

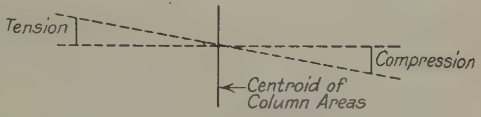


Diagram of Axial Stress in Columns due to Wind.

FIG. 25

As it is usually permissible to ignore stresses due to wind if they are not more than one-quarter of the stresses from dead and live loads, the last calculation is rarely necessary, but for stability the joints should be capable of resisting the bending moments shown in the former analysis.

Wind Load on Bridges. In bridges, the leeward girder is to a certain extent shielded by the windward girder. Thus, in experiments made by Sir Benjamin Baker on lattice girders used in the Forth Bridge, the resistance offered by two girders at a distance apart equal to their depth was about 1·2 that of one girder. With a distance apart 2, 3, or 4 times the depth, the resistance was about 1·5, 1·7, and 1·8 times, respectively, that of one girder. The single lattice girder gave a resistance 1·15 times that of a plane surface of the same dimensions. The presence of an extra girder half way between them increased the wind effect only slightly.

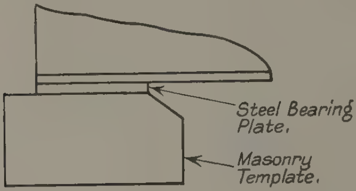


FIG. 26

B.E.S.A., No. 153, requires, however, that if the distance apart of the two girders exceeds twice the depth, the whole exposed surface of each girder shall be taken into account.

The wind load on a train passing over a bridge will transfer stresses to the structure regarded as a horizontal girder, similar to those produced in the vertical girder by a rolling load.

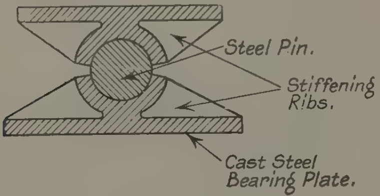


FIG. 27

The wind load on the structure itself produces stresses in the horizontal girder, similar to the dead load stresses in the vertical girder.

The worst overturning effect of the wind on a bridge will occur when the leeward track is covered with the largest empty vehicles that can come on it. In Mr. Adam Hunter's *Bridge and Structural Engineer's Handbook*, the centre of the train is given as 7 ft. 6 in. above the rail level, its area 10 sq. ft. per foot run, and the lightest load 10 cwt. per foot run.

For road bridges the centre is 6 ft. above the road, area 8 sq. ft., and load 6 cwt. a foot run, except for trams for which the corresponding figures are 8 ft., 14 sq. ft., and 10 cwt. per foot run respectively.

FIG. 28

Side Thrust on Bridges due to Track Curvature. When a load such as a train travels in a curve, it exerts a centrifugal thrust which causes a side load on the bridge carrying the track which may act at the same time and in the same direction as the wind. If $W =$ the weight of the train per foot run, $V =$ its speed in miles per hour, and $r =$ the radius of curvature of the track in feet, then

$$\text{the side thrust per foot run} = \frac{WV^2}{15r}$$

which may be taken as acting 5 ft. above the rail level.

Braking Effect. The effect of the sudden application of continuous brakes when a train is crossing a bridge causes a longitudinal force which must be transferred by the structure to the supports. The tractive effort of the engine has a similar effect, and either may be taken as 20 per cent of the weight of the train.

Temperature. In a steel bar an increase in temperature of 1° F. causes its length to increase by about ·0000066 of its original length.

Thus, a range of 100° F. would cause in 100 ft. of steel a change of length equalling $\cdot0000066 \times 100 \times 100 \times 12 = \cdot8$ in. The range observed in the Forth Bridge is less than $\frac{1}{2}$ in. in 100 ft.

For long lengths of steelwork in buildings provision is made for expansion and contraction by slotting the holes in bolted connections, but many large buildings have been erected with no provision for expansion, the stresses set up by the restraint of natural movement being covered by the factor of safety.

End Bearings for Bridges. For girders less than about 70 ft. span, it is usual to rivet a bearing plate under the ends with rivets countersunk on the underside.

The load from this may be transmitted to a masonry bed-plate direct or through sheet lead or layers of felt to make the pressure on the masonry more uniform.

The allowable pressures due to dead load, live load, impact, and centrifugal force are given in B.E.S.A., No. 153, as 25 tons per sq. ft. on granite, 20 on sandstone, 18 on 4–1 cement concrete, and 15 on 6–1 cement concrete. These figures when compared with 600 lb. per sq. in. (38·6 tons per sq. ft.) which is a commonly allowed compressive stress in 1 : 2 : 4 ballast concrete are seen to be not excessive.

When the girder deflects there is a tendency to increase the pressure on the edge of the masonry nearest the opening, and it is advisable to chamfer the edge of the template as indicated in Fig. 26.

At one end, the bearing plate should be held in position by bolts bedded in the masonry. At the other end the holes for the bolts are sometimes slotted to allow slight longitudinal movement, but where it is necessary to make provision for such expansion a better method is to provide for movement between the bearing plate and a steel plate bolted to the masonry; lateral movement being prevented by a projection in one plate sliding in a groove in the other.

For long spans it is common practice for one bearing to come on a large diameter circular bar or pin between two castings, one fixed to the underside of the girder and one bolted to the masonry, as indicated by the cross-section shown in Fig. 27. This arrangement permits of angular movement at the ends, due to the deflection of the girder, without altering the uniformity of the pressure on the masonry. At the other end a similar bearing is provided, but the lower casting instead of being fixed to the masonry rests on rollers with axes perpendicular to the line of the girder which, in turn, bear on another casting bolted to the masonry.

As the movement is not great, the rollers, instead of being completely circular, are often made as in Fig. 28, to permit closer spacing.

The allowable load on rolled steel rollers in tons per inch run is given in B.E.S.A., No. 153, as $d/4$ where d is the diameter in inches which should not be less than 4 in.

Friction. The resistance to end movement will vary with the vertical load on the bearing, and the nature of the surfaces in contact.

According to B.E.S.A., No. 153, for roller bearings the longitudinal thrust to cause movement may be taken as 3 per cent of the vertical load. For sliding bearings of steel on cast iron, the frictional coefficient is 25 per cent.

The longitudinal force to overcome this friction must result in stresses throughout the chord to which the bearing plates are fixed, and will be tensile or compressive according as the temperature is falling or rising.

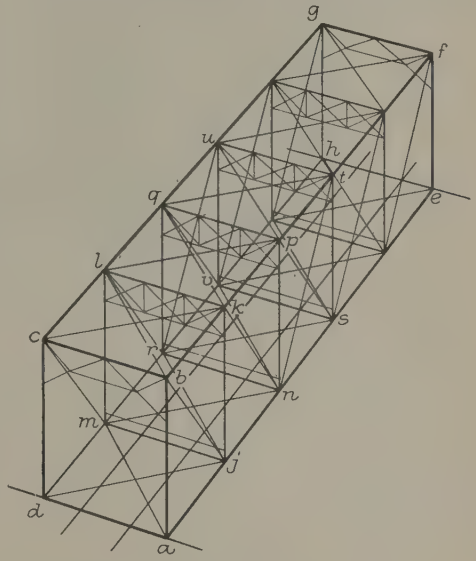


FIG. 29

Braced Girder Bridges. Fig. 29 represents in outline a braced N or Pratt truss for a single line railway.

The members ae and dh are the bottom *booms* or *chords* (tension members); bf and cg are the top booms (compression members); ab and cd are the *end posts*; bj, cm, etc., are *diagonal ties*; jk, np, st, etc., are *vertical posts*. The central diagonals ps, nt are without stress for dead load only, but are subject to alternating stresses as the train crosses the bridge; they are usually made to resist compression.

The lines ad and eh indicate the abutments; jm, nr, etc., are the *cross girders* or *floor beams* carrying the *stringers or rail bearers* parallel to the bottom booms. The ends m, j, etc., of these girders are usually rigidly connected to the vertical posts, stiffening them as do the braced girders at the top of the posts, which form the struts of the wind bracing of which the diagonals ck, bl, etc., are in compression or tension, according to the direction of the wind. The wind load carried by the top booms is partly transferred to the floor system by the frames *jklm*, etc., and partly by the horizontal girder *bfgc* to the ends *bc* and *fg*, and from thence by the frames *abcd*, *efgh*, which have a stiff *portal bracing* at the top.

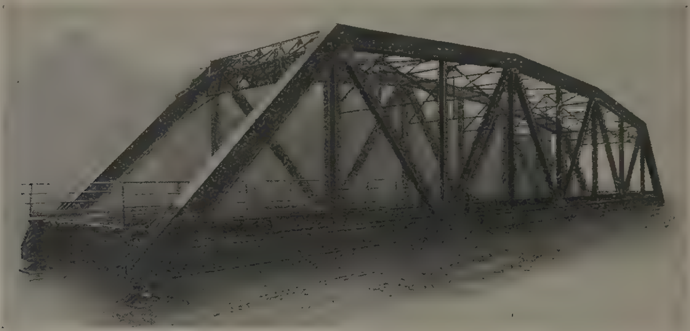


FIG. 30. WARREN TYPE GIRDER, ASSEMBLED IN THE SHOPS

The floor of the bridge is also diagonally braced for wind unless it is made rigid, as when the floor system is carried continuously on the bottom booms.

The half-tone illustrations, Figs. 30 and 31, from photographs kindly lent by Messrs. Dorman, Long & Co., Ltd., show a through girder of the *Warren* type assembled in their shops, and a deck girder only partially assembled.

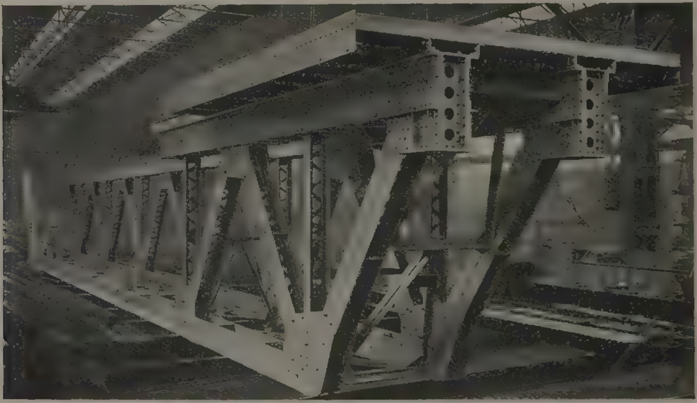


FIG. 31. DECK GIRDER PARTIALLY ASSEMBLED

It will be noted in the through girder that the heavy end posts are inclined, as also are the top booms. In Fig. 29 a heavy compression member could have been inserted between *a* and *k*, and the members *ab*, *bk*, and *bj* omitted with a resulting economy. The sloping of the member *kp* would have reduced the stress in the member *kn*.

The rocker bearings of the type shown in Fig. 27 are clearly illustrated in Fig. 30, as is also the wind and portal bracing.

The deck girder, Fig. 31, has a floor on the top boom of troughing similar to that shown in Fig. 6.

Columns and struts will be discussed in the next lesson.

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XXIV

Search for Reference. Although the relationship between the various sections of engineering is very intimate, the field is so wide that it not infrequently happens that a draughtsman is faced with the problem of designing an unfamiliar machine detail. Unfamiliar, that is, in the sense that the drawing office possesses neither data nor guidance drawing for reference.

In such cases it is not unusual to seek a specialist in the manufacture of this particular machine detail, and give this part of the work to an outside contractor. The alternative is to relate recognized general principles in design to any information which may be available in catalogues or engineering periodicals.

Worm Gearing. As an example of meeting the above contingency we may deal with the following specification of a worm gear.

Twisting moment to be transmitted to the wheel shaft = 20,000 lb.-in.

Speed of wheel shaft = 40 rev. per minute.
Speed of worm shaft = 360 ,, ,,
Service: continuous for 30 minutes.

It is evident that the foundation of the design consists in the diameters of the worm and wheel, and that the dimensions of same will, in turn, be influenced by the nature of the materials used in the construction. Now although precise data may be lacking regarding the outline and proportion of worm gearing, there is a certain amount of information of a general character in circulation, such as the choice of materials to form suitable rubbing surfaces. Tapping this information, we may choose manganese bronze and steel as the most suitable materials for worm gearing, making the worm of steel and the worm wheel of manganese bronze.

Skeleton Design. We have remarked that, given a good reference drawing, the art of machine design consists in a great measure of a combination of interpolation and a building up of existing machine details. Without the reference drawing a different method is necessary. In the latter case we suggest commencing with a skeleton drawing of the part, and using the outline as a guidance in suggesting the various stresses and other considerations which determine the final proportions of the detail.

It cannot be too strongly urged that a rapidly drawn preliminary outline of the machine detail is worth the labour, and invariably saves time and trouble when the final drawing is made.

The above outline drawing is made to no particular scale, but simply to suit the eye regarding a probable proportion. Referring to Fig. 1, we have

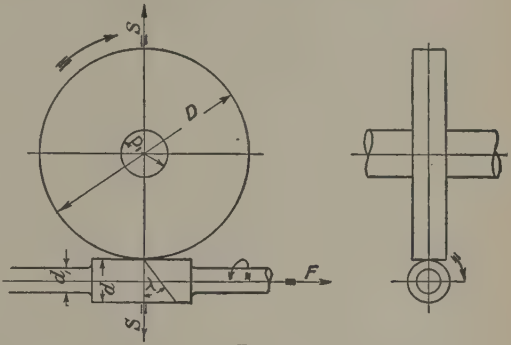


Fig. 1

D = pitch circle diameter of worm wheel in inches (nominal).
d = pitch circle diameter of worm in inches (nominal).
D_1 = diameter of worm wheel shaft in inches.
d_1 = diameter of worm shaft in inches.
F = axial thrust on worm shaft in lb.
S = separating force between worm and wheel in lb.
p = pitch of worm and wheel teeth in inches.
L = lead of worm in inches.
T = number of teeth in wheel.
T_q = twisting moment transmitted through wheel shaft in lb.-in.
t_q = twisting moment transmitted through worm shaft in lb.-in.
f_w = width of wheel teeth in inches.

In designing the gear we may approach the problem along two distinct lines, the one leading to a minimum size and the other to a maximum efficiency. With certain ratios of gear and powers to be transmitted, a maximum efficiency may be obtained within the minimum dimensions but, where a minimum size gear fails to provide a maximum efficiency, a compromise is usually made between the two.

Referring to Fig. I and considering the problem first with a view to finding the smallest size of gear consistent with a reasonable strength, we may relate the various dimensions to the safe working stresses and the power transmitted.

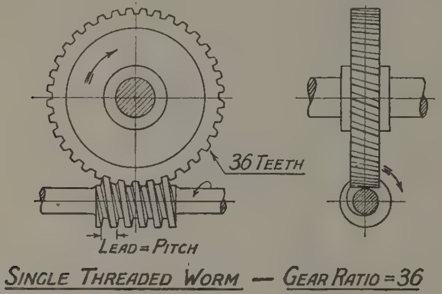


FIG. 2

Finding a suitable value for T, we observe that to fit a single threaded worm as shown in Fig. 2 the number of teeth in the wheel would equal the gear ratio, i.e. 9. The generating of such a small number of teeth would result in undercutting. Where, to avoid undercutting, the number of wheel teeth is made greater than the gear ratio it becomes necessary to make the worm multiple threaded as in Fig. 3, which shows a double-threaded worm.

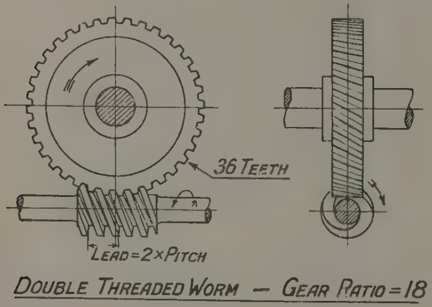


FIG. 3

Considering 30 as a minimum value for T, a gear ratio of 9 would require a *quadruple-threaded* worm giving a value of $T = 4 \times 9 = 36$.

Where, as in the above design, a number of the factors are variables, it is advisable to relate the same in an approximate manner to obtain a working value for one factor which may serve as a foundation in the design. Proceeding to find a suitable value for p we have where

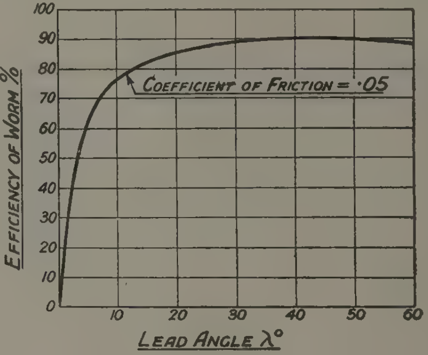


FIG. 4

$B.M.$ = bending moment on tooth in lb-in.
$R.M.$ = resisting moment of tooth in lb.-in.
= stress induced in teeth in lb per sq. in. assuming the force F to be taken by one tooth.

$$B.M. = F \times {\cdot}3683 \,.\, p = \frac{2T_q}{D} \times {\cdot}3683p$$

$$R.M. = \frac{({\cdot}5p)^2 fw \times \text{stress}}{6} \text{ approx.}$$

making $\lambda = 45°$ and $f_w = \frac{3}{4}d$

$$\frac{4p}{\pi d} = \tan 45° = 1$$

$$d = \frac{4p}{\pi} \text{ and } f_w = \frac{3}{4} \times \frac{4p}{\pi} = \frac{3p}{\pi}$$

and allowing a safe working stress of 8000 lb. sq. in.

$$R.M. = \frac{{\cdot}25p^2}{6} \times \frac{3p}{\pi} \times 8000 = 318{\cdot}4p^3$$

$$\text{also } p = \frac{\pi D}{T} \text{ or } D = \frac{Tp}{\pi} = \frac{36}{\pi}p$$

$$\text{then } B.M. = \frac{2T_q \,.\, \pi}{36p} \times {\cdot}3683p$$

$$= \frac{2 \times 20{,}000\pi \times {\cdot}3683}{36}$$

$$= 1285 \text{ lb.-in.}$$

Equating *R.M.* = *B.M.*

$$318{\cdot}4p^3 = 1285$$

$$p = \left(\frac{1285}{318{\cdot}4}\right)^{\frac{1}{3}} = 1{\cdot}592 \text{ in. say } 1\tfrac{5}{8} \text{ in.}$$

$$D = \frac{T.p}{\pi} = \frac{36 \times 1{\cdot}625}{\pi} = 18{\cdot}62 \text{ in}$$

$$F = \frac{20{,}000 \times 2}{18{\cdot}62} = 2148 \text{ lb.}$$

$$S = F . \tan 14\tfrac{1}{2}° = 2148 \times {\cdot}2586 = 556 \text{ lb.}$$

Proceeding to the design of the worm, allowing a safe shear stress in the worm shaft of 6000 lb. per sq. in. and assuming a gear efficiency of 90 per cent.

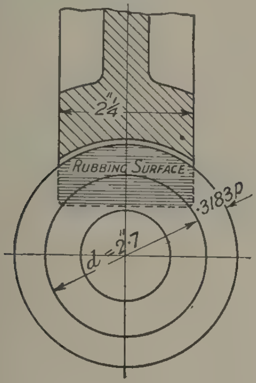


FIG. 5

$$t_q = \frac{20{,}000}{9 \times {\cdot}9} = 2470$$

$$d_1 = \left(\frac{16 \times 2470}{6{,}000\pi}\right)^{\frac{1}{3}} = 2{\cdot}1^{\frac{1}{3}} = 1{\cdot}28 \text{ in. say } 1{\cdot}5 \text{ in.}$$

$$d = d_1 + 2 \times {\cdot}3683p = 1{\cdot}5 + {\cdot}7366 \times 1{\cdot}625$$

$$d = 1{\cdot}5 + 1{\cdot}2 = 2{\cdot}7 \text{ in.}$$

$$\lambda = \overline{\tan}\,\frac{4 \times 1{\cdot}625}{2{\cdot}7\pi} = {\cdot}765 = \tan 37° - 20{\cdot}5$$

Having determined a minimum size of gear in terms of strength we must, before finally accepting the same, consider the condition of the rubbing surfaces.

Let *P* = pressure between worm and wheel in lb. per sq. in.
V = peripheral speed of worm in ft. per sec.
A = area of rubbing surfaces in sq. in.

Referring to Fig. 5, *A* may be approximately estimated to equal 2·1.

$$P = \frac{F}{A} = \frac{2148}{2{\cdot}1} = 1023$$

$$V = \frac{(2{\cdot}7 + {\cdot}3183p \times 2)\,\pi \times 360}{12 \times 60} = \frac{3{\cdot}73\pi}{2}$$

$$= 5{\cdot}857$$

$$PV^{\frac{1}{3}} = 1023 \times 5{\cdot}857^{\frac{1}{3}} = 1850.$$

Although in estimating the above value of *P* we have assumed the whole of the tooth load to be taken by one tooth, the above value of $PV^{\frac{1}{3}}$ is rather high, and we shall be advised to consider the question of increasing the area *A*. This may be done by an increase in diameter *d*, but such an alteration will result in a reduced efficiency of gear.

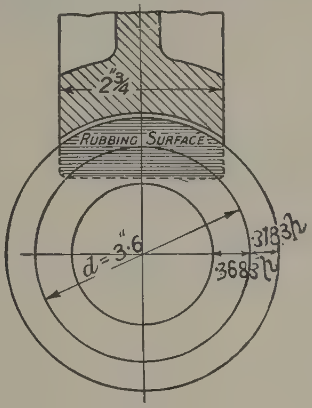


FIG. 6

Assuming a value of 0·05 for the coefficient of friction between the worm and wheel.

$$\text{Efficiency} = \frac{\tan \lambda.}{\tan (\lambda + \phi)}$$

Plotting values of efficiency in terms of the angle λ we obtain a curve as shown in Fig. 4, from which we observe that for values of λ above 30° the increase in efficiency is very small.

Proceeding, we may decide to take a value of $\lambda = 30°$, and so find modified values of *d*, *P*, and *V*.

$$d = \frac{4p}{\pi \tan 30°} = \frac{4 \times 1{\cdot}625}{{\cdot}5773\,\pi} = 3{\cdot}584 \text{ in. say } 3{\cdot}6 \text{ in.}$$

$A = 2{\cdot}35$ approx. (referring to Fig. 6)

$$P = \frac{F}{A} = \frac{2148}{2·35} = 914 \text{ lb.}$$

$$V = \frac{(3·6 + 2 \times ·3183p)\,\pi \times 360}{12 \times 60} = \frac{4·63\pi}{2}$$

$$= 7·27.$$

$$PV^{\frac{1}{3}} = 914 \times 7·27^{\frac{1}{3}} = 1770.$$

Standard Modules. It will be observed that the values of *D* and *d* in the above gear have been determined upon the assumption that the older method of choosing a circular pitch of even dimension would prove the most suitable to the shops in making the part. We may, in passing, briefly consider the modern method, of fixing the diameters *D* and *d*, referred to on page 700, in Lesson VII on *Gearing*.

Proceeding to find the module for the above determined values of *D* and *d*, we have

M = wheel module
q = worm module.

$$M = \frac{p}{\pi} = \frac{1·625}{\pi} = ·5172$$

$$q = \frac{d}{M} = \frac{3·6}{·5172} = 6·96.$$

These values are odd and it will be evident that commencing the design with regard to standard modules would result in slight variation to the above determined values of *D* and *d*.

We may also observe that it is a practice with many designers to base the teeth dimensions, for lead angles exceeding 20°, upon the normal instead of the axial pitch as in the above example.

PIONEERS OF ENGINEERING

By J. F. CORRIGAN, M.Sc., A.I.C.

LESSON XXIII

THE FATHER OF THE CHEMICAL INDUSTRY

JAMES MUSPRATT (1793–1886)

CHEMICAL manufacture in the eighteenth century was a severely restricted occupation. The demand for chemicals was not great, the variety of them was few, and their manufacture was an affair of small scale pot-and-pan processes. But with the ever widening sphere of engineering and industrial interests, and the enormous strides which the science of that day was making, the demand for supplies of chemicals of one variety or another naturally began to increase.

Among the early pioneers who attempted and eventually succeeded in meeting this growing demand in a satisfactory manner was James Muspratt. Muspratt is little known at the present day, yet our modern chemical engineering industries have had their rise in his early efforts. He is, in fact, the "Father of the Chemical Industry."

Born in Dublin, of English parents, in 1793, Muspratt, at a very early stage of his life, showed a decided leaning for pursuits of a chemical nature. He went to a local school, and was afterwards apprenticed to a Mr. Mitcheltree, a Dublin druggist. Here he became acquainted with the business which he was destined to develop and expand during his later years.

Muspratt's first essay in chemical manufacture occurred when he made an endeavour to meet the growing demand for the heavier chemicals. This was in the year 1806. He made hydrochloric acid, potassium ferrocyanide, and similar chemicals on a small scale. The attempt was a successful one, but the manufacture of the chemicals was a difficult task for young Muspratt.

A few years later, Muspratt came to England,

JAMES MUSPRATT

and entered into partnership with a Mr. Josias Christopher Gamble. Together, they erected a small works at St. Helens in 1828. Alkalis formed the chief product of the works. The partnership only lasted for a couple of years, however, after which Muspratt founded new works, and launched out on his own account to manufacture soda by the newly devised Leblanc process. In this manner the British alkali industry was set up.

Muspratt married in 1818. He had ten children, four of his sons—James Sheridan, Richard, Frederick, and Edmund Knowles Muspratt—subsequently becoming chemists and chemical manufacturers of note, and succeeding to the business.

Besides conducting his manufacturing operations according to the recognized processes of the day, Muspratt devised new manufacturing schemes. He laid down the foundations of that, until recently, very much neglected branch of engineering—chemical engineering—devising new principles involving the heating of liquids, transport of liquids from vessel to vessel, distilling operations, and so forth.

Musprattt died on the 4th May, 1886, at Seaforth, near Liverpool, in his ninety-fourth year. Throughout his strenuous career he gained the respect and admiration of his fellow industrialists; and if his manufacturing operations and processes succeeded in transforming rural areas of countryside into ugly and smoke-laden manufacturing districts, the nation benefited thereby, and it was enabled to attain its one time status of the world's leading chemical manufactory.

AN EARLY CHEMICAL EVAPORATING VAT
(From an Old Print)

CHEMICAL CRYSTALLIZING PANS IN THE TIME OF MUSPRATT
(From an Old Print)

GAUGING AND INSPECTION

By Joseph G. Horner, A.M.I.Mech.E.

LESSON V

THE MEASUREMENT OF ANGLES

Bevel Protractors. The setting out and checking of angles or bevels is provided for in several instruments. The common bevel is adjusted from a plain protractor, divided round into degrees. These are made of steel for shop use.

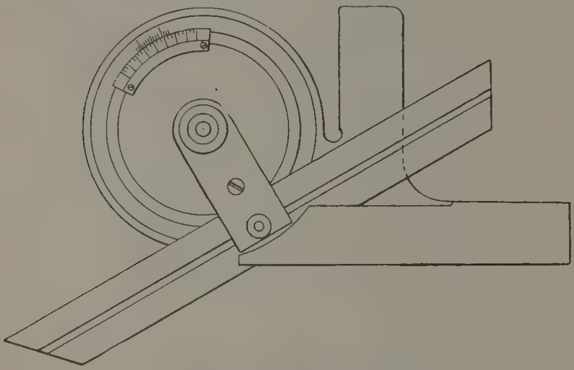


Fig. 30. A Bevel Protractor

An instrument with very extensive utilities is the form of bevel protractor shown by Fig. 30. It comprises a blade, either 6 or 12 in. long, which can be slid along its entire length and clamped, and can be set to any angle relatively to the base by reference to the dial, around which it is moved, and clamped with the central stud. The dial is graduated in degrees round the entire circle, the lines being sunk slightly below the surface. A vernier is fitted which reads to 5 min., or $\frac{1}{12}^\circ$. The method of reading is shown in the enlarged diagram, Fig. 31. Each space on the vernier is 5 min. shorter than two spaces on the main scale. When the 0 line on the vernier coincides with the 0 line on the main scale, the edges of the base and blade are parallel. When the swivel head is moved so that the line on the vernier next to 0 coincides with the line next but one to 0 on the main scale, the angle included between the base and the blade is 5 min. or $\frac{1}{12}^\circ$. Hence the rule, read off directly from the main scale the number of whole degrees between 0 on the main scale and 0 on the vernier. Then count in the same direction the number of spaces from the 0 of the vernier scale to a line that coincides with a division on the main scale. Multiply this number by 5, and the product will be the number of minutes to be added to the whole number of degrees. In Fig. 31 the 0 line on the vernier has moved 12 whole degrees from the 0 line on the main scale, and the eighth line on the vernier coincides with a line upon the main scale, as indicated by the arrow. Multiplying 8 by 5, the product, 40, is the number of minutes to be added to the degrees, 12°, 40 min. The instrument can be used to check angles in all conceivable positions and varied conditions.

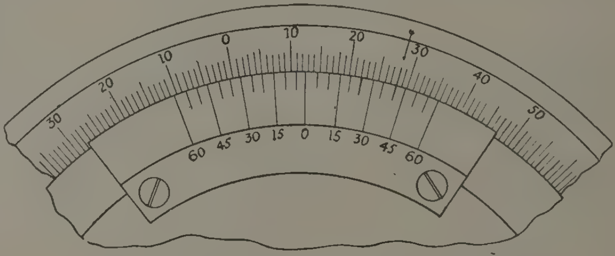


Fig. 31. A Bevel Protractor with Vernier

A protractor of a different type for laying down or checking angles is shown by Fig. 32. The blade, 6 in. long, is locked with a nut. The graduations are in degrees from 0 to 180, both ways. The rectangular head gives four working faces. A similar instrument has a semicircular head. Another useful design is illustrated in Fig. 33. A rule fully divided passes through an adjustable circular turret which is graduated from 0 to 180°, to read both ways. The head extends on both sides of the blade, so that angles can be transferred from either side without resetting. These are made with English and metric graduations, with rules of different lengths, and in combination forms that include the turret or dial, a square, and a centre square.

The Sine Bar. This is one of the later instruments for measuring angles, the claim being made that it is more reliable than setting angles with the protractor, the disc of which is of small diameter. It is based on a trigonometrical function of an angle, namely, the sine, which is the relation which the height bears to the hypotenuse of a right-angled triangle, that is, $\frac{AB}{CB}$. Values are obtained directly from a table of sines.

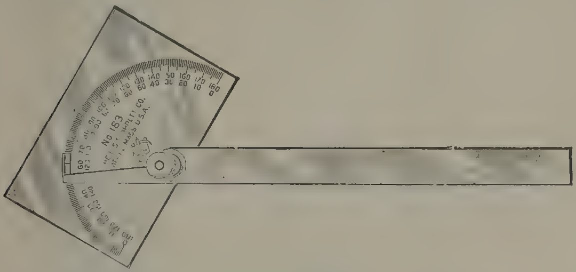


Fig. 32. A Bevel Protractor Giving Four Working Faces

In its essentials, the sine bar (Fig. 34) is a strip of steel having two hardened studs set at precise centres, generally 10 or 5 in. in the axis of the strip, which give a sufficiently long basis for measurement. Slots are provided for clamping it against the face of an angle plate or a slotted upright. The 10 in. bar is convenient, because, since the functions of angles are based on a radius of 1, if the decimal in the table of sines is moved one place to the right this is equivalent to multiplying by 10. If a bar with 5 in. centres is used, then the sine is multiplied by 5.

The height of the bar can be set with any of the standard gauge blocks, or height gauges. Thus, taking a bar with plugs at 10 in. centres, to set the bar to an angle of 30°, 10 min., the sine is 0·50251 in. Moving the decimal one place to the right gives 5·0251 in. from the centre vertically, of the lower pin to that of the upper, plus the distance of the lower pin from the table. Measurement is more conveniently taken from the lower edges of the studs than from their centres. Instead of clamping, the angle plate may be drilled and bushed to locate the left-hand end of the sine bar at either of the holes with a plug. Then the height to the right-hand stud is measured and set, plus that of the height to the plugged hole.

The sine bar mounted on an angle plate is not the most convenient form, since it is not entirely self-contained. A better design is the Taft-Peirce, Fig. 35, in which the bar is of an angular form like a try-square, pivoted on a stud in the angle, the stud being adjustable vertically, and clamped in a slot in an upright member mounted on a base. A parallel straight-edge is set on the base from which all datum measurements are taken. The lower edge of the sine bar, and the top of the straight-edge are used for checking angles. The centres of the studs are 5 in. If the bar is to be set for an angle of 2°, 23 min., the sine of this is found to be 0·04159. Multiplying this by 5 (for 5 in. centres) the result is 0·20795, which is the distance the movable stud must be set higher than the pivot stud. Sine bars are useful for checking the angles of bevel wheel blanks. Two bars mounted on a plate and set, test taper gauge and standard tapers.

Angle Gauges. The Johansson system has been extended in one direction to measure angles by the combinations of angle blocks. The single angles are formed at the ends of strips. There are 85 pieces in three series. In one, comprising 15 pieces, the angles increase in minutes from 10 to 11°. In another, with 40 pieces, angles are included from 0 to 90° in increments of 1°. In the third, with 30 pieces, a range is covered from 89 to 90° in minutes, so that each angle is larger by 1 min. than the previous one. The pieces in each series have from two to four angles each.

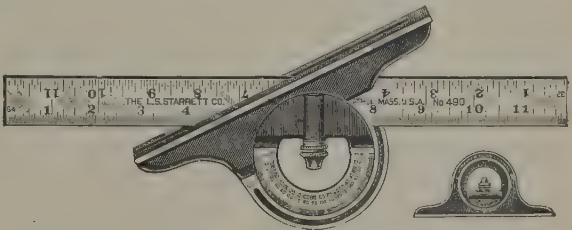


Fig. 33. Protractor with which Angles can be Transferred from either Side

The Three-wire System. Fixed screw thread gauges, outside and internal, are used generally for checking commercial screw threads. But more refined methods are in use when more accurate results are required, as in the gauges themselves and in some instruments. In one of these—the three-wire method, Fig. 36—two wires of gauged diameters are laid in contact with the sides of the thread on one side, and one on the opposite side, coming midway between the two. Then a micrometric measurement is taken over the wire, and a calculation made with a formula that varies with different sections of thread. In Fig. 36, D is the diameter over the wires, d that over the tops of the thread, P the pitch, and W the diameter of the wires.

Then $D = d - AP + BW$,
Then for Whitworth threads,
$A = 1{\cdot}6008$ and $B = 3{\cdot}1657$,
and for Sellers threads,
$A = 1{\cdot}5156$ and $B = 3{\cdot}000$.

When this method of measurement is adopted, the screw profile must be exactly standard.

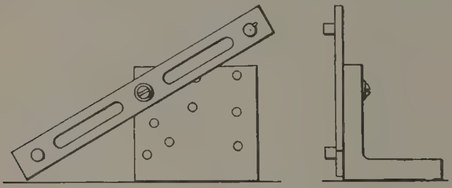


Fig. 34. A Sine Bar

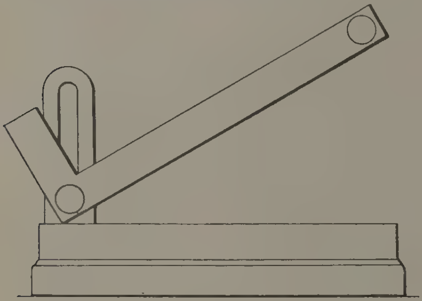


Fig. 35. The Taft-Peirce Sine Bar

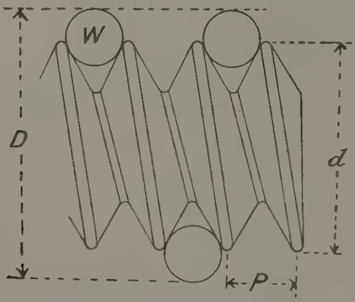


Fig. 36. Three-wire Method of Measuring Screw Threads

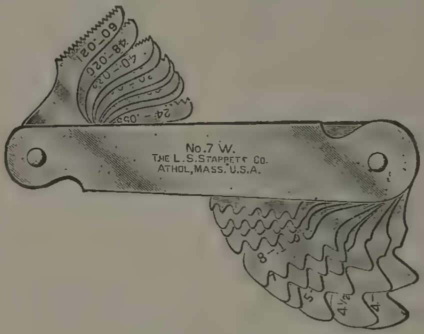


Fig. 37. Whitworth Screw Pitch Gauge with 26 Pitches

The Leaf Gauges. These are a very large group of fixed gauges, comprising an assemblage of thin steel leaves, having the edges shaped to suit the objects to be tested. The most common are those for measuring the pitches of screws, Fig. 37. Each group includes a number of threads of many pitches of one standard size and proportion, Whitworth, Pipe, Metric. The serrations on the edges of the leaves correspond

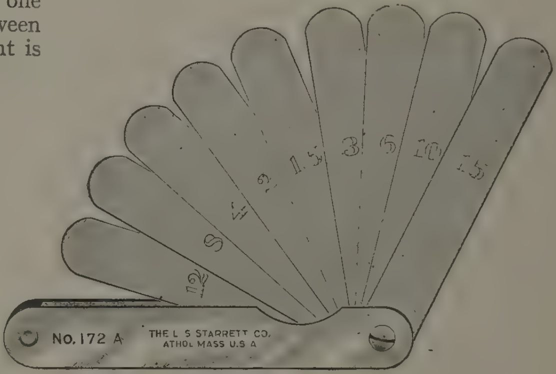


Fig. 38. Thickness Gauge with Nine Leaves

with the standard thread sections, from 22 to 30 in number. When the teeth that correspond with the thread to be tested are ascertained, the pitch can be read on the leaf. The end of each leaf is narrow to permit of passing it within a nut. To enable the workman to find the sizes of drills for holes to be tapped, each leaf is stamped to show the double depth of thread, whence the size is obtained for a full thread. If the thread is not to be of full depth, an allowance will be made.

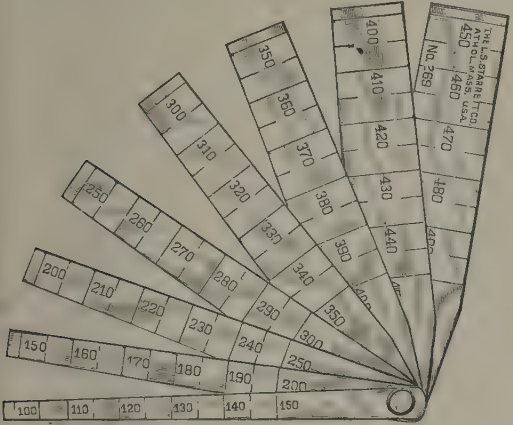


FIG. 39. A TAPER GAUGE FOR MEASURING HOLES

The same design of leaves is embodied in many other gauges. Some of these measure the radii of fillets for forming tools and dies. In another the ends are ground to angles in degrees, from 1 to 45°. Others, of which Fig. 38 is typical, measure the thicknesses of minute slots and spaces that cannot be tested by other means. They are stamped for different systems. Each leaf may be used singly or in combination with others. Tapered gauges are a variety of the leaf designs, Fig. 39. They are graduated to read in thousandths of an inch. They measure the width of slots, and the inside sizes of tubes and holes. Calipers can also be set to size, within the range. A combined taper, thickness, and wire gauge is shown by Fig. 40, which shows the front, and reverse sides of the two leaves, one for wire, the other as a rule, on one side, and on the other giving thicknesses corresponding with tapers.

The wire gauges with notches on circular or oblong plates are familiar. They are used for wire and sheets. They lack uniformity in numbering, having evolved like screw threads from crude beginnings. The Birmingham or Stubs gauge is that used in England, but for precise dimensioning it is necessary to state the sizes in decimal parts of an inch that correspond with gauge numbers.

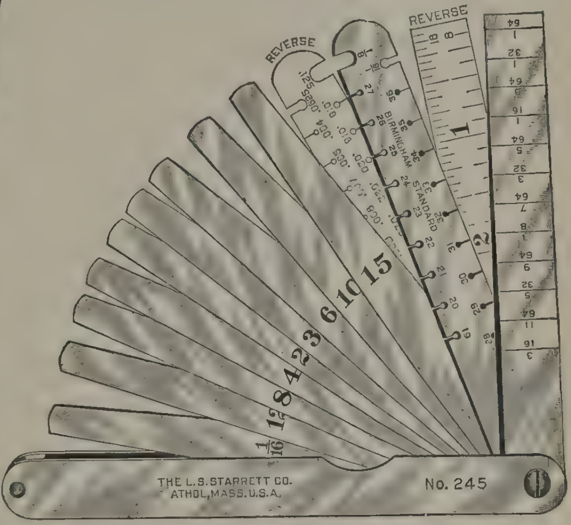


FIG. 40. A COMBINED TAPER, WIRE, AND THICKNESS GAUGE

(CONCLUSION)

BLAST FURNACE PRACTICE

By Percy C. R. Kingscott, D.I.C., A.R.C.Sc., F.I.C., B.Sc. (Lond.)

LESSON X

ORES

Treatment of Ores

Ideal furnace practice is based upon uniformity, one of the many factors being the grading of the ores. To this end, it has become the custom to crush the ores to definite sizes, depending upon the nature of the material and the practice adopted. Grading of material has a great influence upon the distribution of the burden column, thus ensuring that most essential uniform contact with the ascending gases. A common size is that of 4 in., i.e. the ore being crushed so that all the pieces will pass through a screen with mesh of that size.

Apart from this operation, many iron bearing substances require special treatment, the different beneficiation processes being described briefly below.

Roasting consists in heating the ore in contact with air, without causing sintering, in order to remove sulphur.

$$FeS + 2O_2 = FeSO_4$$
$$4\,FeSO_4 = 2\,Fe_2O_3 + 4\,SO_2 + O_2$$

This operation is generally conducted in kilns where the heating is accomplished by burning solid, liquid, or gaseous fuel. One of the commonest types of kiln is the Gjers kiln, which is a vertical steel cylindrical shell provided with a bosh, and lined with firebrick. The height is usually 20 to 35 ft., and the diameter 15 to 25 ft. The bottom of the shell is provided with suitable openings for the removal of the roasted ore and for the air inlets. Ore is charged at the top and spread at intervals with a layer of fuel which may be slack coal or coke breeze, the operation of the kiln being continuous.

A well-known type of gas-fired kiln is the Davis-Colby, in which the ore descends in a comparatively thin sheet between firebrick walls, being subjected to the action of burning gases. A recent modification of this type consists of a long rectangular kiln in which the roasting chambers are built in duplicate. There is a firebrick chamber on each side of each of the two ore chambers, gases being burnt in the outside chambers which communicate with the inner ores by suitable openings. The innermost chambers collect the products of combustion and connect to the chimney. Roasted ore is removed by chutes at the bottom.

Calcining is also performed in kilns as described above, but is undertaken for a different reason, viz., in order to remove water, to decompose carbonates, to render dense ore more porous, or to alter the state of oxidation of the ore to the magnetic oxide in order to facilitate subsequent concentration.

Carbonates, such as the clay ironstone of Cleveland, when calcined, lose carbon dioxide according to the equation—

$$4\,Fe\,CO_3 + O_2 = 2\,Fe_2O_3 + 4\,CO_2$$

The operation renders the ore porous and in a form readily accessible to the reducing gases of the blast furnace. As the reaction is endothermic, it is more economically achieved in a kiln than in the furnace itself.

Magnetites can be pulverized, or changed to a higher form of oxide Fe_2O_3, by suitable treatment. On the other hand, "hematites" can be converted into the magnetic oxide in a reducing atmosphere, preferably at a low temperature, in order to avoid fusion of the ore.

$$6\,Fe_2O_3 = 4\,Fe_3O_4 + O_2$$

Large lumps and fines should be avoided, the ore being preferably of a uniform size.

Concentration takes the form of washing, jigging or magnetic concentration, the latter being either wet or dry in application.

Washing is applied to the removal of argillaceous material from nodules of ore. Running water removes the clay particles, leaving the heavier ore.

Jigging is applied to sandy ores. The latter are placed in boxes with perforated bottoms, which are caused to pulsate in water. The contents of the boxes are thus agitated and become separated into layers, the fine sand rising to the top and passing away, whereas the heavy ore travels to the bottom.

Magnetic separation is resorted to with ores naturally or artificially magnetic. They are crushed to suitable fineness and passed in a thin layer past magnets which separate the gangue from the magnetic oxide.

Leaching with chemicals is occasionally adopted.

Briquetting consists essentially in the agglomeration of fine ores. The latter are mixed with a binder or merely moistened, pressed into shape under heavy pressure, dried and generally burnt in a kiln, similar in operation to a continuous brick-kiln. Flue dust can be utilized either by mixture with ores as above, or by mixing with a catalytic binder, such as magnesium chloride or ferric sulphate, as in the Schumacher process, which renders burning unnecessary.

Nodulizing, as with briquetting, concerns the treatment, at high temperatures, of fine ores and flue dust, and consists in fusing them, at high temperatures, into lumps in a rotary kiln similar to a cement kiln.

Sintering of fine ores and flue dust is now of great importance, several types of apparatus having been designed.

(*a*) Up-blast converters of the Huntingdon-Heberlein type are similar to a Bessemer steel furnace. They are provided with a bottom grate and positive blower. Ore and fuel are mixed in correct proportions on a fire, and after the " blow " the sinter is poured out on to a grating, where it is broken into suitable sizes.

(*b*) Down-draught processes are exemplified by the Dwight-Lloyd and Greenawalt sintering plants. In both these processes, the raw materials are sintered on a grate, the products of combustion being drawn down through the bed. The former is a continuous operation, the latter being intermittent, one definite charge being dealt with at a time. Briefly, a layer of fine ore is spread over the grate, and this is covered by a definite mixture of ore and fuel. The top surface is ignited by a flame, the down draught drawing the flame through the mass, which is reduced to sinter. After ignition, the kindling flame is removed. The down-draught sinter is porous and well de-sulphurized, being better than the up-draught type, owing to the agitation in the latter operation.

Sinter is an excellent material for the blast furnace, and is claimed to be less refractory than dense ores. At the same time, although porous, it has not the carbon dissolving action of fine ores.

EFFECT IN THE BLAST FURNACE

Magnetites retain their oxygen more tenaciously than other ores, thus requiring extra fuel for reduction, much of the latter, it is thought, being effected by solid carbon in the furnace hearth. The speed of reduction is slower with magnetites than with the sesqui-oxides, again increasing the cost of smelting.

In order to effect the maximum amount of reduction in the stack, i.e. with carbon monoxide, it is essential that the ratio of surface to bulk of the ore should be large, otherwise the ore, e.g. magnetite, reaches the hearth partially reduced, a fruitful source of cold slags.

On the other hand, exceedingly fine ores, although conforming to the requirement relating to surface, unfortunately have a catalytic action causing deposition of carbon in the top of the furnace.

$$2\,CO = CO_2 + C$$

Fine ores have a tendency to run ahead of the rest of the burden and, owing to the intimate contact with the lumps of fuel, cause a loss of fuel by solution loss and solid reduction. They are also productive of heavy loss in flue dust blown from the furnace.

The physical condition of the ore has a great influence on the distribution of the burden in the furnace, as it is clear that lumpy ores will slide off the bell with greater inertia than plastic or fine ores. The consequence of this separation is to deliver the ores in different positions relative to the coke charge, and to the furnace walls, and any changes in the charging of ores is bound to have serious effect upon the working of the process. As has already been shown, ore crushing is resorted to with these objects in view, viz., distribution of the burden and the production of a greater surface in the ore, experience indicating the size of the broken ore to be 3 to 4 in., although the most recent tendency is to break to even smaller size.

Imperfectly calcined ferrous carbonate is the source of considerable loss in furnace practice. It absorbs heat in the hearth, causing the formation of cold slags, and it is considered that the nascent ferrous oxide requires solid carbon for reduction.

LOCALITY

Carbonate ores are worked in quantity in Europe, especially in Great Britain, Germany, Styria, and France. There are large deposits, in America and Canada, of lean carbonate not yet commercially important.

Magnetites are found in England, Sweden, India, America, and Canada.

Hematites and **Brown** ores are found in England, Germany, France, Sweden, Spain, India, America, North Africa, Cuba, Newfoundland, and Canada.

Rich siliceous Iron schist occurs in Brazil.

The chief iron producing districts of **Great Britain** are: Cleveland district yielding Middle lias; Lincolnshire, Rutland, and Northants yielding Inferior oolite; North Lincolnshire yielding Lower lias; South Lincolnshire, Leicestershire, and Oxfordshire yielding Middle lias; Westbury yielding upper Corallion; North West England producing rich low phosphorus hematite; North Staffordshire producing Black band from the carboniferous strata.

Typical analyses of the Jurassic ores, according to Hatch, are as follows—

Locality	Fe	SiO_2	Al_2O_3	CaO	S	P	H_2O
Cleveland	28·1	11·8	10·2	4·7	·26	·47	6·8
Northants, Rutland	32·5	14·7	6·1	2·7	·10	·60	15·2
North Lincolnshire	22·7	8·1	5·1	18·2	·16	·31	10·7
Leicester and South Lincolnshire	25·2	10·9	8·0	9·6	·11	·25	16·4
Oxford	24·0	10·2	7·6	12·2	·06	·23	15·6

A typical analysis of North West coast hematite is—

Fe	SiO_2	CaO	P
48·5	15·5	1·7	·007

ANALYSES OF TYPICAL ORES

Ore	Fe	SiO_2	CaO	Al_2O_3	MgO	P	Mn	S	H_2O
Rubio	44·83	13·90	1·46	2·75	tr.	·018	·79	·037	8·32
El Halia	49·30	6·75	·73	2·51	·33	·021	·46	·13	8·70
Rouina (A)	51·75	8·00	5·46	3·30	—	·02	·55	·054	4·15
Rouina (B)	49·77	11·59	4·12	3·35	·34	·044	·34	·04	4·19
Breira	50·63	4·53	2·52	3·27	·75	·009	1·14	·055	6·58
Dicido	45·44	8·56	1·24	2·21	·30	·024	·80	·018	14·83
Philipville	64·79	2·79	·39	1·47	tr.	·01	·14	·167	1·98
Miliana	48·93	6·60	3·93	1·59	·84	·015	1·28	·021	6·44
Villaricos	42·00	8·02	3·61	—	·41	·019	7·40	·55	5·08
Welsh Manganiferous	5·25	25·00	2·40	10·0	·33	·09	28·0	·05	7·00
Cartagena	25·90	10·84	3·15	4·85	·54	·019	17·08	·40	7·32
Gellivare	62·33	3·02	2·88	—	—	·76	·14	·14	·75
Santander	53·06	3·26	·56	4·66	·33	·047	1·24	·076	6·67
Wabana	53·86	9·50	1·81	3·55	·84	·80	·65	·018	4·30
Mesaba	55·76	7·49	·15	·81	·12	·024	·47	·009	9·00
Alabama	37·00	13·44	16·2	3·18	—	·37	—	·07	·50
Roll Scale	71·00	2·50	—	—	—	—	—	—	—
Cinder	52·00	25·0	·3	1·5	—	·07	·3	—	—

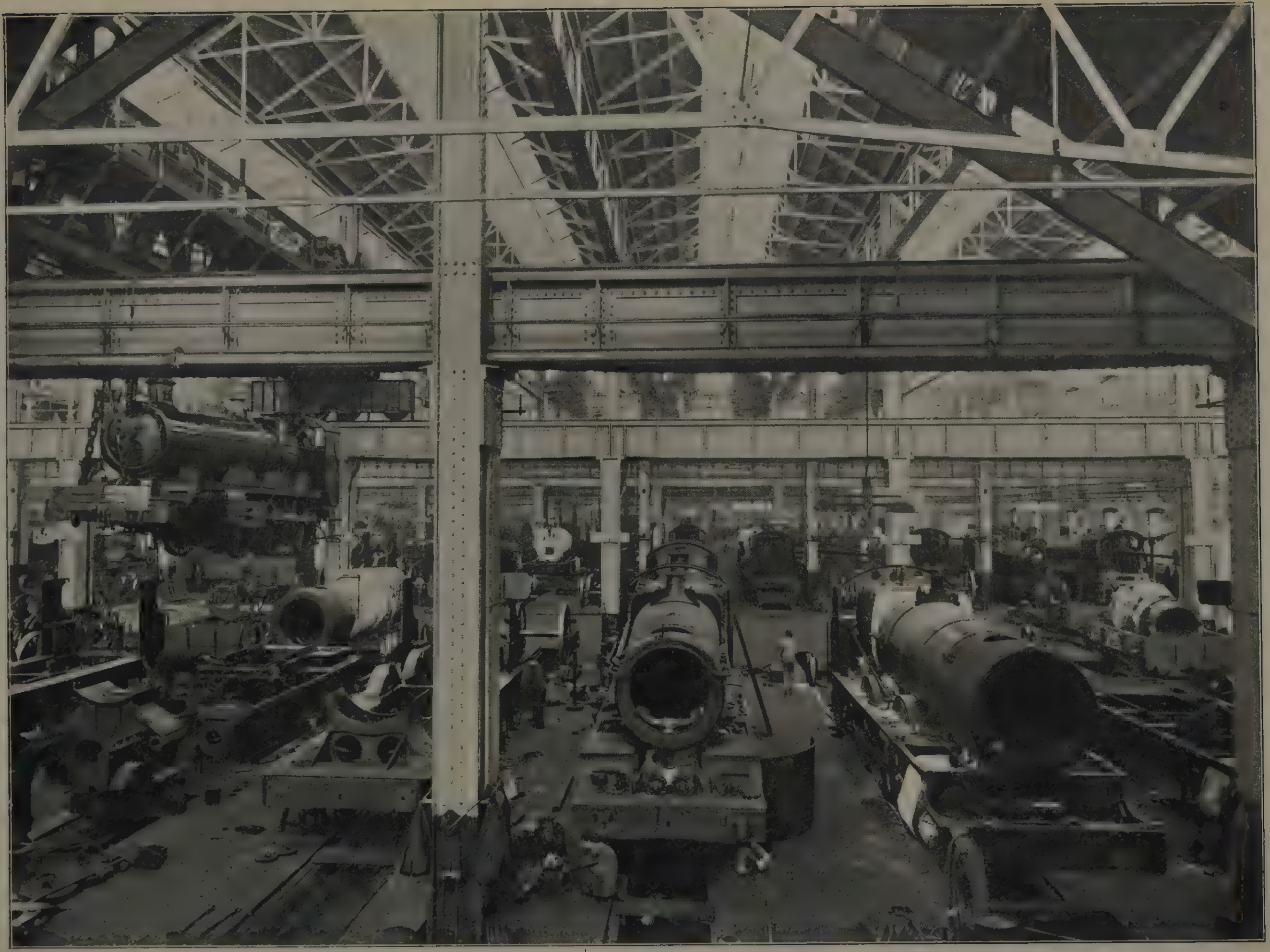


Fig. 81. Erecting Shop, Great Western Railway Works, Swindon

(See Article on "Fitting and Erecting," page 1270)

MATHEMATICS FOR ENGINEERS

By W. G. BICKLEY, M.Sc.
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LESSON XXVI

VOLUMES

ANOTHER common case in which results have to be worked out by integration is the calculation of volumes of solids of varying cross-section. In this case we imagine the solid cut up into a large number of slices by parallel planes, and if A is the area of any slice, and x is the distance of the slice from the end (measured *perpendicular* to the planes of section), the thickness of a slice will be denoted by δx, and its volume (δV) will be given by (area) × (thickness), or in symbols,

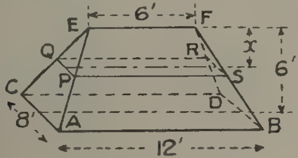


FIG. 101

$$\delta V = A \,.\, \delta x.$$

Just as in finding the area under a curve, we must add up the volumes of the slices to find the total volume, at the same time imagining the slices to become thinner and thinner, the number of slices becoming correspondingly greater; the result of this can be expressed in the same notation, and the process of evaluation carried out in the same manner. Thus

$$V = \int A \,.\, dx$$

the limits being, of course, the values of x at the ends of the solid or portion of a solid we are considering. The relation between A and x (which must, of course, be known before we can carry out the integration) depends upon the shape of the solid, and must usually be found from its geometry.

As a first instance we will find the volume of a pile sketched in Fig. 101, on a rectangular base 12 ft. by 8 ft., the sides sloping up to a ridge 6 ft. long, and the vertical height of the pile being 6 ft. In this case, A will denote the area of a section $PQRS$, at a distance x (measured *vertically*) below the ridge. It will be seen, by proportion, that $\frac{PQ}{AC} = \frac{x}{6}$ or $PQ = 8\,\frac{x}{6}$. Also, PS will be $EF + \frac{x}{6}$ (increase in length from EF to AB) $= 6 + \frac{x}{6} \times 6 = 6 + x$. Therefore $A = PQRS = \frac{8}{6}\,x\,(6 + x) = 8x + \frac{4}{3}\,x^2$. Also the limits of the integral will be 0 and 6, since in going from top to bottom, x increases from 0 to 6. Thus

$$\begin{aligned} V &= \int_0^6 \left(8x + \frac{4}{3}\,x^2\right) dx \\ &= \left[4x^2 + \frac{4}{9}\,x^3\right]_0^6 \\ &= \left(4 \times 6^2 + \frac{4}{9} \times 6^3\right) - (0) \\ &= 240 \text{ cub. ft.} \end{aligned}$$

Solids of Revolution. The majority of the solids with which the engineer has to deal are, however, solids of revolution, in which case the section perpendicular to the axis is a circle.

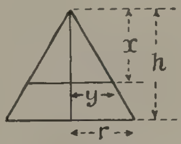


FIG. 102

Then, if we denote by y the radius of the section at distance x (measured along the axis), we shall have $A = \pi y^2$ so that our formula for volume becomes

$$V = \int \pi y^2 \,.\, dx$$

The commonest forms are the cone and the sphere, and we will first work out formulae for the volumes of these.

CONE. In this case we have from the similar triangles in Fig. 102 (which represents an elevation of a cone, with its axis vertical) $\frac{y}{x} = \frac{r}{h}$ or $y = \frac{r}{h}x$. Thus, putting in the limits, evidently 0 and h, we have

$$V = \int_0^h \pi \frac{r^2}{h^2} x^2 \, . \, dx$$
$$= \left[\pi \frac{r^2}{h^2} \frac{x^3}{3} \right]_0^h$$
$$= \left(\pi \frac{r^2}{h^2} \frac{h^3}{3} \right) - (0)$$
$$= \tfrac{1}{3}\pi r^2 h.$$

SPHERE. In this case the elevation is a circle of radius r, and from the triangle OPN (Fig. 103), we have $y^2 = r^2 - x^2$. Also, the extreme values of x are $-r$ and r. Thus

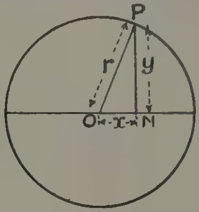


FIG. 103

$$V = \int_r^r \pi (r^2 - x^2) dx$$
$$= \left[\pi (r^2 x - \tfrac{1}{3}x^3) \right]_{-r}^{r}$$
$$= \pi (r^3 - \tfrac{1}{3}r^3) - \pi (-r^3 + \tfrac{1}{3}r^3)$$
$$= \frac{4}{3}\pi r^3 = \frac{\pi}{6} d^3$$

(where d is the diameter).

Frustum. The volume of a frustum of a cone may be obtained by subtracting the volume of the cone removed from that of the complete cone, or from the formula for this case, but it is often simpler to obtain the result from first principles, by integration. We will illustrate this by finding the volume of a frustum of a cone, the radii of the ends being 5 in. and 8 in., and the height of the frustum being 6 in. A side view, with the axis horizontal, is given in Fig. 104, and we must first find the formula for the radius y, in terms of x. Here y must be $5 + \frac{x}{6}$ (increase in radius) $= 5 + \frac{x}{6} \times 3 = 5 + \frac{1}{2}x$. Also, the limits are 0 and 6. So

$$V = \int_0^6 \pi (5 + \tfrac{1}{2}x)^2 dx$$
$$= \int_0^6 \pi (25 + 5x + \tfrac{1}{4}x^2) dx$$
$$= \left[\pi \left(25x + \frac{5}{2}x^2 + \frac{1}{12}x^3 \right]_0^6 \right.$$
$$= \pi \left(25 \times 6 + \frac{5}{2} \times 6^2 \right.$$
$$\left. + \frac{1}{12} \times 6^3 \right) - (0)$$
$$= 258\pi = 810{\cdot}5 \text{ cub. in.}$$

Portion of Sphere. Similarly, if we require the volume of the portion of a sphere of radius 9 in., cut off by parallel planes 5 in. and 8 in. from the centre, and on the same side of it, we shall have $y^2 = 81 - x^2$ (Fig. 103, with $r = 9$) and our limits will be 5 and 8. So

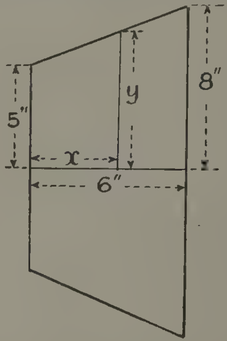


FIG. 104

$$V = \int_5^8 \pi (81 - x^2) dx$$
$$= \left[\pi (81x - \tfrac{1}{3}x^3) \right]_5^8$$
$$= \pi (81 \times 8 - \tfrac{1}{3} \times 8^3) - \pi(81 \times 5 - \tfrac{1}{3} \times 5^3)$$
$$= 114\pi = 358{\cdot}2 \text{ cub. in.}$$

MOMENTS OF INERTIA

The term moment of inertia is used in engineering to denote two different types of quantity. One, as when used in the mechanics of rotating bodies, is a true moment of *inertia*, being the sum of the masses times the squares of their distances from the axis, and the unit in which this is usually measured is *lb.-ft.*^{2} The other has nothing to do with *inertia* at all, properly speaking, for it relates to a section of a beam undergoing bending or twisting. It is very closely related mathematically, however, for to find it we multiply each small *area* (instead of *mass*) by the square of its distance from the axis (neutral axis in bending or axis of twist in twisting), and take the sum for all the small areas into which the total area is divided. The unit is usually *ins.*^{4}. Since the square of the distance comes in in both cases, the methods of finding, and also the rules for changing axes, or for finding "moments of inertia" of compound bodies or areas, will be the same, so that it is advantageous to denote them by the same term. Strictly speaking, it would be better to use the term "Second Moment" of mass or area as the case may be, and this is sometimes done—the "second" referring to the power to which the distance is raised. As we obtain these moments in all cases by adding, and must divide up the mass or area into small portions, it will be seen that to find second moments, integration must usually be employed, as we shall soon show. There is another quantity which is important in this connection, however, and that is "radius of gyration." Primarily referring to rotating bodies, it is there defined as the length by the square of which the total mass must be multiplied to give the moment of inertia, i.e.

$$I = Mk^2.$$

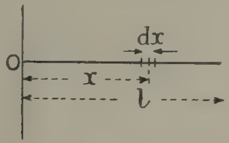


FIG. 105

For the other case, we replace mass by area, and then have

$$I = Ak^2,$$

"k" thus depends upon upon the shape, and it is useful to remember k for the commoner shapes.

Typical Moments of Inertia. 1. ROD. We will evaluate the moment of inertia of a rod about an axis perpendicular to it, through one end. If l is the length and M the mass, then the mass of a small piece dx at a distance x from the end (see Fig. 105) is $\frac{M}{l}dx$. We must multiply this by the square of its distance from the axis (i.e. by x^2), and integrate from $x = 0$ to $x = l$. Thus

$$\begin{aligned} I &= \int_0^l \frac{M}{l}\,x^2 dx \\ &= \left[\frac{M}{l}\,\frac{x^3}{3}\right]_0^l \\ &= \frac{M}{l}\,\frac{l^3}{3} - 0 = \tfrac{1}{3}Ml^2. \end{aligned}$$

We see also that in this case, $k^2 = \frac{1}{3}l^2$.

The student should verify that if we want the moment about an axis perpendicular to the rod through its centre, we must integrate the same expression between $-\frac{1}{2}l$ and $+\frac{1}{2}l$, obtaining $I = \frac{Ml^2}{12}$, and $k^2 = \frac{l^2}{12}$.

RECTANGLE. In dealing with the bending of beams, the moment of inertia of a rectangle about an axis through its centre parallel to one side will often be required. In this case (see Fig. 106) we divide the area into strips parallel to the axis, and with the dimensions there given, the area of the strip is $B\,.\,dx$. We must multiply this by x^2 and integrate from $-\frac{1}{2}D$ to $+\frac{1}{2}D$. So

$$\begin{aligned} I &= \int_{-\frac{1}{2}D}^{\frac{1}{2}D} Bx^2\,dx \\ &= \Big[\tfrac{1}{3}Bx^3\Big]_{\frac{1}{2}D}^{\frac{1}{2}D} \\ &= \left(\frac{B}{3}\,\frac{D^3}{8}\right) - \left(-\frac{B}{3}\,\frac{D^3}{8}\right) \\ &= \frac{BD^3}{12} \end{aligned}$$

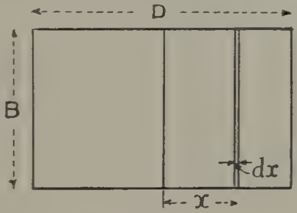


FIG. 106

Also, since the area is BD, we have $k^2 = \frac{D^2}{12}$, a result which should be compared with that for a rod about an axis through its mid-point perpendicular to it.

Circle. In the torsion of a circular shaft, we shall require the moment for a circle about an axis perpendicular to its plane through its centre. We divide the circle up into concentric rings (see Fig. 107). The area of a ring of radius x and width dx is $2\pi x \,.\, dx$; multiply this by the square of its distance from the axis (x^2) and integrate from 0 to r.

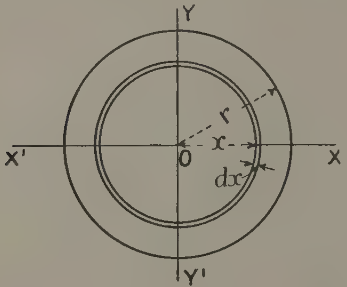


Fig. 107

$$\begin{aligned} I &= \int_0^r 2\pi x^3\,dx \\ &= \left[\tfrac{1}{2}\pi x^4\right]_0^r \\ &= (\tfrac{1}{2}\pi r^4) - (0) \\ &= \tfrac{1}{2}\pi r^4 = \frac{\pi d^4}{32} \end{aligned}$$

if d is the diameter. As the area is πr^2, we deduce that $k^2 = \frac{1}{2}r^2$. This will also be the value of k^2 for a disc fly-wheel, so that for it, if M is the mass, we shall have $I = \frac{1}{2}Mr^2$. In the bending of a bar of circular cross-section, we shall need the moment about a diameter, XOX^1. By symmetry this is the same as that about YOY^1, and it is known that the sum of these is the moment of inertia about the axis perpendicular to the plane that we have just found. Thus, about a diameter,

$$I = \tfrac{1}{4}\pi r^4, \text{ and } k^2 = \tfrac{1}{4}r^2.$$

It is impossible in the space to devote more attention to this subject, but other properties of moments of inertia will be found in the articles on Applied Mechanics.

CENTRE OF GRAVITY

Another class of "addition" problems that require integration is that of finding the centre of gravity of bodies or areas. The fundamental formula is that, if x is the distance of the centre of gravity from an axis, and the body can be divided into small portions $m_1, m_2 \ldots$ at distances $x_1, x_2 \ldots$ from the axis,

$$\bar{x} = \frac{\text{sum of moments}}{\text{sum of masses}} = \frac{\Sigma mx}{\Sigma m}$$

These sums will usually be obtained by integration. We will give a few instances.

Triangle. We will find the distance of the centre of gravity from an axis (YOY^1 in Fig. 108) through the vertex, parallel to the base.

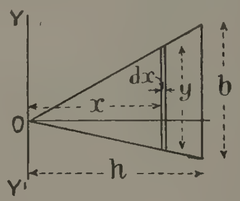


Fig. 108

Divide the area into strips parallel (to the base. Then, if y is the length of one of these, distant x from the axis, and dx is its width, the area is $y\,.\,dx$. The sum of the "masses" (areas here) will be the area of the triangle, $\frac{1}{2}bh$. For the sum of the moments we shall have $\int_0^h xy\,.\,dx$. To find this, we need a formula for y in terms of x. By similar triangles, we have $y/b = x/h$, or $y = \frac{b}{h}x$. Thus the sum of the moments will be

$$\int_0^h \frac{b}{h}\,x^2\,dx = \left[\frac{b}{h}\,\frac{x^3}{3}\right]_0^h = \tfrac{1}{3}bh^2$$

On performing the division, we find $\bar{x} = 2h/3$.

Cone. By symmetry, the centre of gravity of a cone is on its axis, so we only need to know the distance from the vertex. If y is the radius of the cone at a distance x from the vertex (measured along the axis), the volume of a slice of thickness dx is $\pi y^2 dx$, and this corresponds to "m." So Σm is the volume of the cone, or $\frac{1}{3}\pi r^2 h$, r being the radius of the base, and h the perpendicular height. The moment of the slice will be $\pi y^2 x\,.\,dx$, and this must be integrated between 0 and h. Here $y = \frac{r}{h} x$ (see above where the volume was calculated), so that

$$\sum mx = \int_0^h \pi \frac{r^2}{h^2} x^3 \, dx = \pi \left[\frac{r^2}{h^2} \frac{x^4}{4} \right]_0^h = \tfrac{1}{4}\pi r^2 h^2$$

By division, we find that $\bar{x} = \frac{3}{4}h$.

Area under Curve. If we want the position of the centre of gravity of the area under a curve, and know its equation, we divide it into strips, and then do the "additions" by integration. As an instance we will find the position of the centre of gravity of the area under $y = 2 + x$, from $x = 0$ to $x = 4$. The area is $OABC$ in Fig. 109. The area of the strip is $y \,.\, dx$ ("m" in the fundamental formula) so

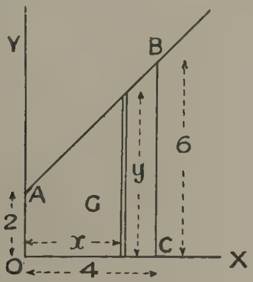


FIG. 109

$$\sum m = \int_0^4 y \,.\, dx = \int_0^4 (2 + x)\, dx$$

$$= \left[2x + \tfrac{1}{2}x^2 \right]_0^4 = 16$$

$$\sum mx = \int_0^4 xy \,.\, dx = \int_0^4 (2x + x^2)\, dx$$

$$= \left[x^2 + \tfrac{1}{3}x^3 \right]_0^4 = 37\tfrac{1}{3}$$

So $\bar{x} = 37\frac{1}{3} \div 16 = 2\frac{1}{3}$.

To find Σmy, we note that the centre of gravity of the *strip* is half-way up, and so the moment about the x axis is $\frac{1}{2}y \times y \,.\, dx = \frac{1}{2}y^2 dx$.

$$\sum my = \int_0^4 \tfrac{1}{2}y^2 dx = \int_0^4 (2 + 2x + \tfrac{1}{2}x^2) dx$$

$$= \left[2x + x^2 + \tfrac{1}{6}x^3 \right]_0^4 = \frac{104}{3}$$

Dividing by Σm, we have $\bar{y} = \frac{13}{6}$. The position of the centre of gravity may now be plotted.

It will be seen from the method of working this last example that we can write the formulae for finding the position of the centre of gravity.

$$\bar{x} = \frac{\int xy \,.\, dx}{\int y \,.\, dx}, \quad \bar{y} = \frac{\int \frac{1}{2} y^2 dx}{\int y \,.\, dx}$$

EXERCISE No. 40

1. A reservoir has a square bottom of side 50 ft., and the sides slope out at 45°. Show that the area at a depth x ft. is $(50 + 2x)^2$ sq. ft., and deduce by integration the volume, if the depth is 30 ft.

2. The curve $y = 2\sqrt{x}$ is rotated about the x axis. Find the volume of the solid from $x = 0$ to $x = 4$.

3. Find, by integration, the volume of a frustum of a cone, radii 2 in. and 5 in., and height 6 in.

4. A bowl is a portion of a sphere of 12 in. radius, and its depth is 6 in. Find its volume.

5. Show that the radius of gyration of a ring is given by $k^2 = \frac{1}{2}(R^2 + r^2)$, R and r being the outer and inner radii respectively. Find the moment of inertia of a fly-wheel rim, 4 in. square in section, inner radius 10 in., if the material weighs 480 lb./cub. ft.

6. Show that the centre of gravity of a hemisphere is at $\frac{3}{8}r$ from the centre, r being the radius.

7. Find the position of the centre of gravity of the area under the curve $y = 2\sqrt{x}$, from $x = 0$ to $x = 4$.

ANSWERS TO EXERCISE No. 39

1. (*a*) 64; (*b*) 28; (*c*) $14\frac{2}{3}$; (*d*) $5\frac{1}{3}$; (*e*) 0; (*f*) 15; (*g*) 20; (*h*) 227·0.
2. 33.
5. $466\frac{2}{3}$; 518·5 cub. yd.
6. 2411.
7. 4189.

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XXV

Use of the Catalogue. Proceeding with the worm gear design, we may now search for a reference among the catalogues of special details to be found in a set of worm gearing. Such special machine parts as ball and roller bearings form a common feature of worm gear design, and a catalogue providing illustrations of the application of such bearings may include the kind of gear we are particularly interested in.

Fig. 1 shows an application of roller and ball bearings to worm gearing, and is taken from a catalogue issued by The Hoffmann Manufacturing Co., Ltd. This catalogue also provides dimensions of bearings suitable for various loads and speeds of rotation and, given the magnitude of the load to be taken by the bearing, one can readily choose a suitable type.

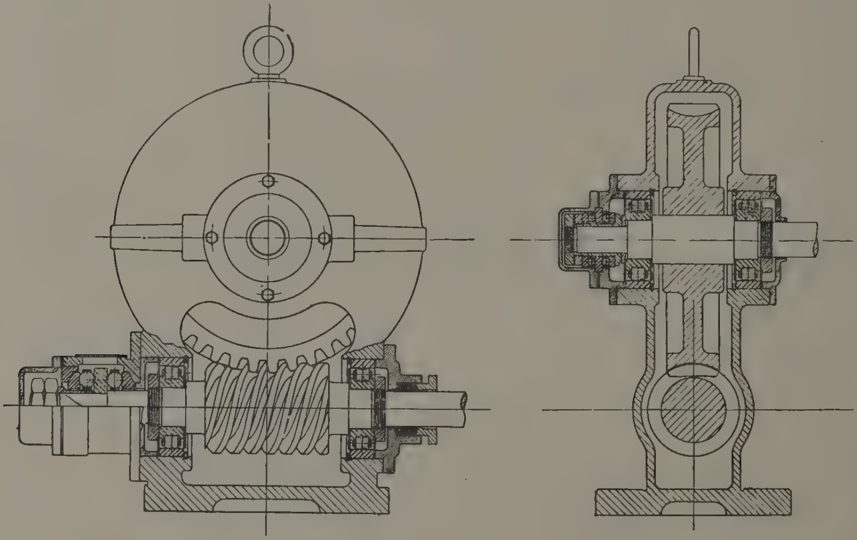


Fig. 1

Deciding to use Fig. 1 as a reference drawing, we may proceed to determine the loads to be taken by the journal and thrust bearings. The proportion of the friction angle to the worm angle is such that, to simplify the analysis, we may neglect friction and proceed to estimate the probable loads as follows. Referring to Fig. 2

Let F = axial thrust on worm shaft in lb.
S = separating force between worm and wheel in lb.
W = tangential force exerted by worm at pitch line in lb.
w = bearing load due to thrust F in lb.
J = bearing load due to thrust W in lb.
t_q = twisting moment transmitted through worm shaft in lb.-in.

It will be observed that the force F is resisted, by the thrust bearing, at the worm shaft axis. Consequently, a couple is produced equal to $F \times d/2$, which is balanced by the resistance of the worm shaft journal bearings Nos. 3 and 4 producing a balancing moment of $14.w$.

$$\text{then} \quad w = \frac{F \times 1{\cdot}8}{14} = 276 \text{ lb.}$$

$$\text{also} \quad W = \frac{t_q}{\frac{d}{2}} = \frac{2470}{1{\cdot}8} = 1372 \text{ lb.}$$

The resistance to the above force W is taken by the wheel thrust bearing at the wheel shaft axis, and the couple produced is equal to $W \times \frac{D}{2}$, which produces a load on each of the wheel shaft bearings of J lb.

$$J \times 7 = W \times \frac{D}{2}$$

$$J = \frac{1372 \times 9{\cdot}31}{7} = 1825 \text{ lb.}$$

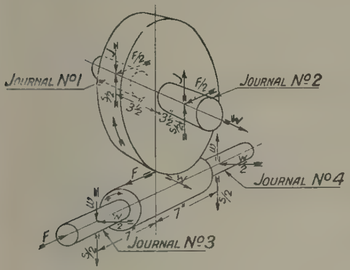


FIG. 2

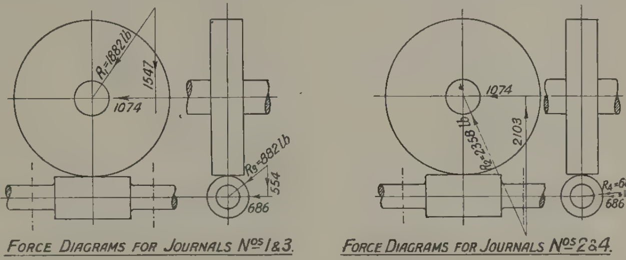

R₁=1882 lb
1547
1074
R₃=882 lb
554
686
FORCE DIAGRAMS FOR JOURNALS Nºˢ 1 & 3.
1074
2103
R₂=2358 lb
R₄=686 lb
686
FORCE DIAGRAMS FOR JOURNALS Nºˢ 2 & 4.

FIG. 3

We may now proceed, as in Fig. 3, to draw force diagrams to determine the resultant forces on the journals.

We are now in a position to choose a suitable type of bearing to take each of the above loads. The maker's catalogue, referred to above, gives the maximum load at various speeds of rotation, which enables the designer to choose quickly a safe type. Finally, when ordering the thrust and journal bearings, it is advisable to specify the load together with the condition of service for each bearing and obtain confirmation regarding the suitability of the choice.

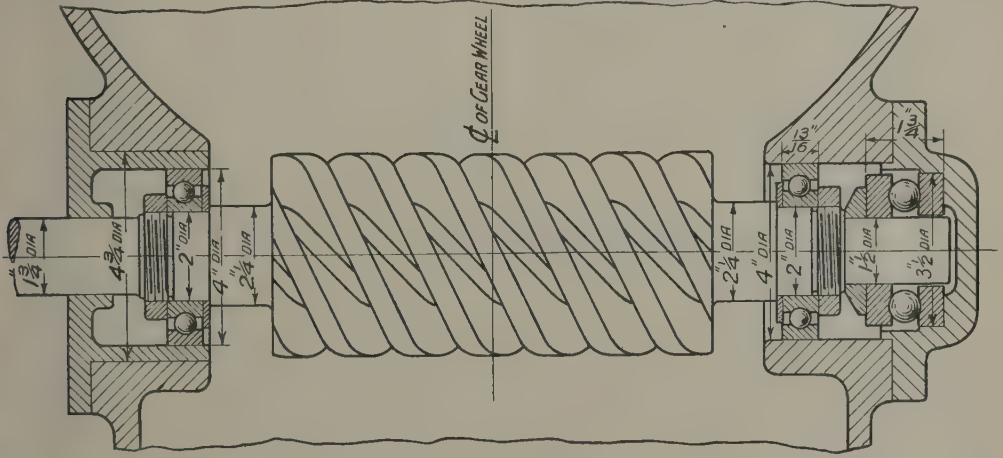


FIG. 4

We have already remarked that the reference drawing is to be used as a general guidance rather than a literal model. Referring to Fig. 1, we observe that the thrust bearing is designed to take axial loads in each direction which is necessary in a reversible gear. Where, as in the above example, the gear is designed to run in one direction only, the axial loads, in both worm and wheel shafts, will act in one direction, and a single thrust washer will suffice in each case. This departure from the reference drawing will enable us to simplify the housing for the washers, and a suggested modification to the casing is shown in Fig. 4 which also gives a detail of the worm shaft.

Interchangeability. Where standard parts are fitted to a machine detail, as in the case of the above bearings, it is often possible to fit the same type throughout the part, and we shall be advised to consider the possibility of fitting a common type of bearing to the above gear.

The loads taken by the worm wheel journals are greater than those taken by the worm shaft journals, but the speed of rotation of the former is only about 11 per cent of the worm speed and, if the diameter of the wheel shaft will allow, we may fit the same type to each shaft. Allowing the same value of shear stress in the two shafts

$$D_1 = \left(\frac{16 \times 20{,}000}{6000\pi}\right)^{\frac{1}{3}} = 2{\cdot}57 \text{ in.}$$

This minimum diameter of wheel shaft excludes the possibility of using a common type of ball bearing to the two shafts.

Turning to the thrust bearing, we have found that a heavy type is necessary for the worm shaft, but for the wheel thrust, which is of smaller value at a lower speed of rotation, we may use the light type shown in Fig. 5, and noting the difference in cost between the heavy and light types makes the choice of the heavier type for the purpose of interchangeability impracticable.

Heat Generated. Before deciding the extent to which we may follow Fig. 1 as a guidance in making a working drawing of the worm gear casing, we ought to satisfy ourselves regarding the capacity of the gear to deal with the heat generated by the gear in overcoming the frictional resistances of the moving parts.

Let H = heat generated by worm in B.Th.U. per minute.

η = efficiency of worm.

$$\text{then } \eta = \frac{\tan \lambda}{\tan (\lambda + \phi)} = \frac{\tan 30^\circ}{\tan (30^\circ + 2{\cdot}86^\circ)}$$

$$\eta = \frac{{\cdot}5774}{{\cdot}6459} = {\cdot}894$$

$$H = \frac{2\pi \,.\, t_q}{12} \times \frac{40 \times 9}{778} \times (1 - \eta)$$

$$H = \frac{2\pi \times 2470}{12} \times \frac{360}{778} \times (1 - 1{\cdot}894)$$

H = 63·48 B.Th.U.'s per minute.

Heat generated in the specified thirty minutes

$$= 63{\cdot}48 \times 30 = 1904{\cdot}4 \text{ B.Th.U.'s}$$

Allowing a safe temperature rise of 100° F. in the lubricating oil and assuming a specific heat of 0·5. Then, if all the heat is absorbed by the oil, the minimum quantity required

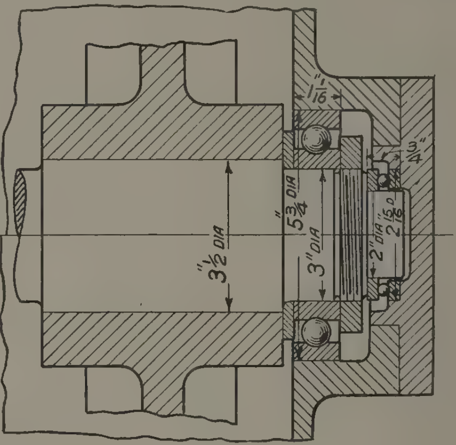


Fig. 5

$$Q = \frac{1904{\cdot}4}{{\cdot}5 \times 100} = 38{\cdot}1 \text{ lb.}$$

$$= \frac{38{\cdot}1 \times 6{\cdot}23}{59} = 4 \text{ gal.}$$

A rough estimate of the capacity of the lower part of the worm gear case will confirm that the quantity of lubricating oil normally present in the case is sufficient to absorb the heat generated in friction without raising the temperature of the oil above the safe limit. We may thus ignore the less definite value of heat dissipated by radiation.

EXERCISE

Referring to Lessons X and XX, make a detail drawing of the worm wheel and shaft, to suit the particulars fixed in the above calculations and the journal and thrust bearings shown in Fig. 4 and 5. Next, using Fig. 1 as a guidance, make a working drawing of the worm gear casing. Finally, find the stresses induced, when transmitting the specified twisting moment, in the worm and wheel teeth.

GAS ENGINES AND PRODUCERS

By Arnold Rimmer, B.Eng.

LESSON XI

GAS ENGINE PERFORMANCE, etc.

The performance and efficiency of a gas engine are determined by the carrying out of certain tests, the nature and extent of which will depend on whether they are conducted from a commercial or a scientific standpoint.

In a commercial test the important points are—

1. Horse-power developed.
2. Fuel consumption.
3. "Governing" of engine at various loads.
4. Amount of overload that can be carried for a definite time.
5. Valve setting.
6. Quantity of cooling water and lubricating oil required per brake-horse-power hour.

A complete scientific test would also be concerned with the performance of the engine as a heat engine, and so would include measurements of the heat supplied in the fuel and that carried away by the cooling water, exhaust gases, etc.

Power and Efficiency. The power developed in the cylinder is calculated from the indicator cards and is, of course, termed the "indicated horse-power."

If P = "mean effective pressure," in pounds per square inch.
A = area of piston, in square inches.
L = length of stroke, in feet.
E = number of explosions per minute.

then $$\text{i.h.p.} = \frac{P.A.L.E.}{33{,}000}$$

The effective horse-power available at the engine crankshaft is determined by some form of "brake." Thus, ropes carrying a dead load may be applied to the fly-wheel, or a special drum fixed to the same and the power absorbed by the friction between the two (Fig. 67).

If W = dead load, in pounds.
w = spring balance reading in pounds.
D = diameter of brake (to centre of ropes), in feet.
N = revolutions per minute.

then $$\text{b.h.p.} = \frac{(W-w)\,\pi.D.N.}{33{,}000}$$

The heat generated may be absorbed by a circulation of water in a special channel on the inside of the wheel or drum, while on larger engines some form of dynamometer or an electrical load would be necessary.

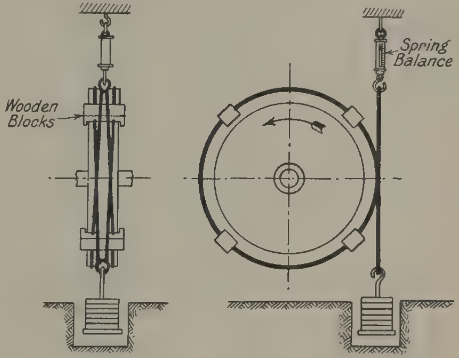


Fig. 67. Rope Brake

The difference between the i.h.p. and b.h.p. represents the power absorbed by the engine itself owing to friction of bearings, piston, etc., together with the "fluid friction" of the gases passing the valves, etc. The "mechanical efficiency" is given by the ratio $\frac{\text{b.h.p.}}{\text{i.h.p.}}$, and will, in general, vary from 70–90 per cent.

Fuel Consumption. The measurement of the consumption of fuel will depend on the nature of the latter. If it is town's gas, it is usually passed through a suitable meter, and for comparative purposes the reading should be reduced to an equivalent consumption at "standard temperature and pressure" (32° F. and 30 in. mercury). In the case of engines running on producer gas it is very usual to state the quantity of coal burnt in the producer per brake-horse-power hour, though in scientific tests the actual gas consumption may be measured by means of a large gas-holder.

Thermal Efficiency. If the performance of the engine as a heat engine is required, it is necessary to determine the heating value of the gas or coal, and this may be effected in some form of calorimeter. Then

$$\text{Thermal efficiency} = \frac{\text{heat converted into work}}{\text{heat supplied to engine}}$$

A complete test would also involve the measurement of the heat carried away by the cooling water and the exhaust gases, together with other calculations which would not be required in an ordinary commercial test.

Valve Settings. As previously pointed out, the proper working of a gas engine is dependent, among other things, on a correct functioning of the valves. Any irregularities in the operation of the latter can be detected from the indicator card and should, of course, be at once adjusted. The valves being operated through cams or similar gear, they have a more or less gradual opening and closing with an intermediate period during which they are full open. Hence, they generally have a "lead" and a "lag" relative to the dead centre positions of the piston and crank. Makers have their own particular values of these "settings," depending on the cam shape, area of valve ports, etc., though, of course, the general conditions existing in the working of the engine do not admit of any great variations in them.

The instant at which the different events of the cycle occur may be determined either in terms of the corresponding crank angle or the piston position. The latter is preferable for comparative purposes, as a crank angle basis needs also particulars of the relative length of the connecting rod. The dead centre position may be determined by means of a spirit-level laid on the crank web, and measurements may be taken between the front of the piston and that of the cylinder liner. The usual method is to have the engine barred round slowly (after first disconnecting the ignition wires), and determine the moment when the rollers on the valve-operating levers come in contact with the cam, as shown by the fact that they can then no longer be turned on their spindle by hand. This gives the instant of valve opening. Similarly, when the rollers become just free again is the instant at which the valve may be assumed to be closed.

The admission valve usually has a "lead," in order to take advantage of the scavenging effect of the previous exhaust stroke, and also to utilize the first outward movement of the piston for drawing the charge into the actual cylinder. Similarly it has a slight "lag," so that the inertia of the gases may be allowed to fill up the cylinder at the end of the stroke. The lag and lead are about equal, or if anything, the former is slightly greater, corresponding to, say, $\frac{1}{2}$ in. of piston movement. If the lead is excessive the pressure existing in the cylinder may prevent induction, and some of the exhaust products pass the admission valve and mix with the new charge. If, on the other hand, the lead is non-existent or "negative" (i.e. the valve opening *after* the dead centre), the engine will be starved, and some of the exhaust gases might be sucked back into the cylinder. Excessive lag causes a delay in the beginning of compression though, of course, the "volumetric efficiency" should be as high as possible.

As previously stated, the gas valve usually opens after the admission valve, and closes before it. This, of course, avoids loss of gas through the exhaust valve, and enables the air to sweep into the cylinder the last portion of gas in the admission valve casing. For engines running on producer gas, the period of operation of the gas valve may correspond exactly with the outstroke of the piston. If, however, the engine is using town's gas which requires a much greater proportion of air, the gas valve may not open until the piston has moved some little distance from the inner dead centre, though it will probably be closed at about the outer dead centre.

In the case of the exhaust valve, lead is required in order that the bulk of the exhaust gases may have left the cylinder by the time the piston reaches the end of its stroke, though if this lead is excessive the valve will have to be lifted against comparatively high pressure with consequent increase of friction and wear of the cam and roller. A slight lag (usually $\frac{1}{8}$ in. or $\frac{1}{4}$ in. of piston movement) is necessary to ensure the "scavenging" of the cylinder is as complete as possible.

Small modifications are usually possible by a screw adjustment on the operating mechanism, though, of course, makers' settings should not be altered without good reason. It should be noted that lag and lead must be considered together since an earlier opening of a valve by this means also involves a later closing and vice versa.

Lubrication. The oil used for the lubrication of gas engines must be of good quality and, preferably, that specially manufactured for the purpose. This applies particularly in the case of the cylinder. Owing to the conditions prevailing there, an oil is required which has a high ignition point and also leaves no deposit after burning.

The general tendency on modern engines is towards forced lubrication, though many of the smaller engines rely on "sight-feed," or on "ring and trough" lubrication for the main bearings, etc., with centrifugal oilers for the crankpin. In some cases, the drip-feed lubricators are replaced by one or more of the plunger type, mechanically operated by the engine and supplying a regulated quantity of oil to the various points. The type used on the Premier multi-cylinder engine was illustrated in Fig. 24, on page 587, while the one fitted on the Crossley 100 b.h.p. engine is shown in greater detail in Fig. 68.

The gudgeon pin is usually supplied by means of a "wiper" fitted at the front of the cylinder, the oil being conducted through channels formed in the top of the piston. (See also Figs. 21, 25, and 28, on pages 586 and 587.) To prevent "blowing back" due to the explosion pressure, the oil to the cylinder is usually fed through an orifice fitted with a non-return valve (generally a steel ball).

The general design of gas engines is based, of course, on mechanical principles, though special consideration is necessary in view of the high pressures and varying temperatures to which many parts are subjected. In addition, there is the possibility of variations in mixture strength producing conditions much in excess of the normal, so that a large "factor of safety" is necessary. Hence, as shown in the descriptions and illustrations of representative engines already given, the trend of modern practice is towards great strength and robustness combined with simplicity of construction, ease of operation, and efficiency and reliability of performance.

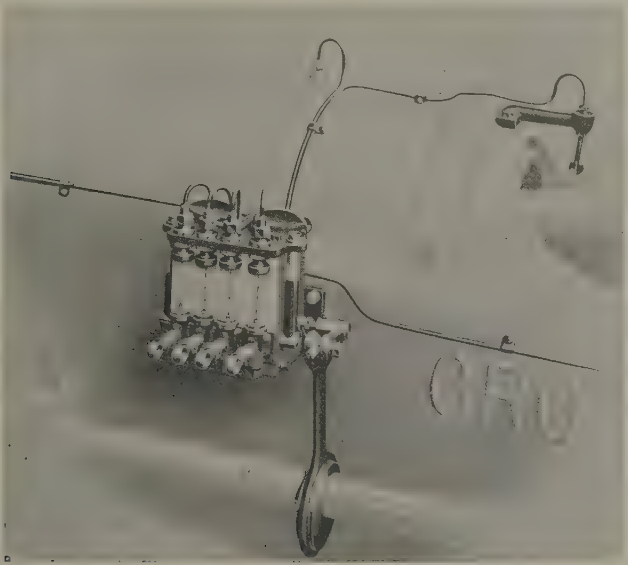


FIG. 68. MECHANICAL LUBRICATOR (CROSSLEY)

PIONEERS OF ENGINEERING

By J. F. Corrigan, M.Sc., A.I.C.

LESSON XXIV

THE INTERNAL COMBUSTION ENGINE DEVELOPED

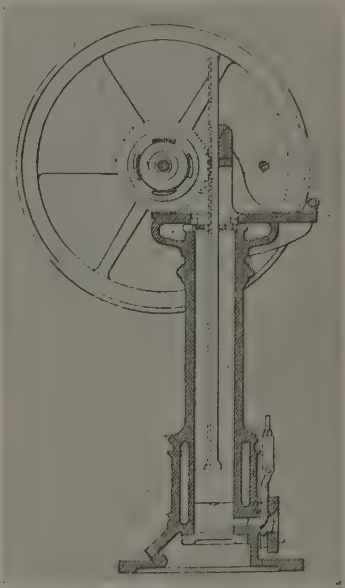
NICOLAS OTTO (1833–1891)

More than any other mechanical power producer since the invention of the steam-engine, has the internal combustion engine been of service to the civilized world. The gas and the petrol engine have provided an alternative means of obtaining power, a means which is all the more welcome in view of the increasing demands which are being made upon the world's coal resources.

Nicolas Otto

Although many experimenters tried their hands at devising an engine which would develop motive power from the explosive combustion of gases, the basis of modern engines of this type is to be found in the pioneer work of Nicolas Otto.

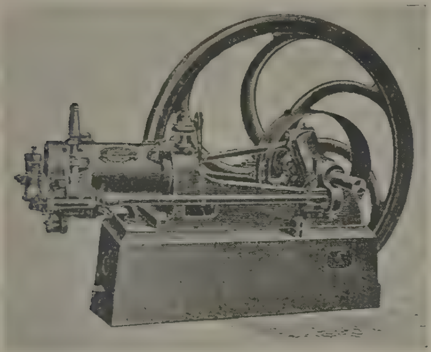
Otto's struggles with the gas engine began in 1854, in which year he commenced his experiments with this type of engine. In 1867 he achieved success in conjunction with his partner, Langen. Otto's early gas engine was a crude contrivance. In the cycle of operations there was no compression of the explosive gas. The mixture was taken into the cylinder during an early part of the upstroke of the piston. The gas was ignited by momentary contact with a flame by means of a specially constructed slide valve. An explosion occurred, after which the piston rose to the top of its stroke, and then descended, chiefly in virtue of the pressure of the atmosphere on top of it. Rackwork gearing was provided, and the engine was excessively noisy and violent in its action.



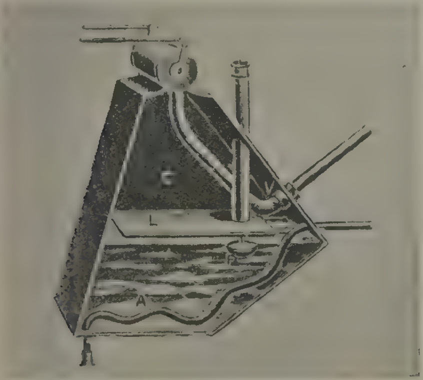
Otto's First Gas Engine, 1867

In 1876, Otto was successful with an improved type of engine. He halved the consumption of gas, and, by means of other extensive improvements and modifications, he devised an engine which quickly rose to the status of a practicable engineering, industrial, and commercial proposition. Ten years later, there were over 30,000 of these engines in use.

Otto was born in 1833. He received only a very ordinary amount of education, and practically no scientific training at all. During the earlier period of his life he was a commercial traveller. His undoubted mechanical interests and abilities, however, ultimately asserted themselves, with the result that he managed to acquire the necessary amount of time and finance to undertake the long series of experiments with gas engines which were necessary before success was granted to him.

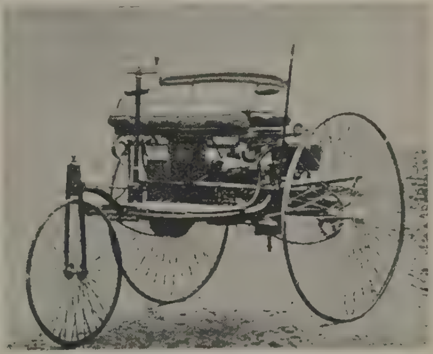


An Otto Gas Engine made by Crossleys in 1880



An Early Carburettor
(From a Contemporary Diagram)

The success of Otto and his co-workers induced others to follow the subject. Otto's engine was of the four-stroke type. In 1881, however, Dugald Clerk invented the two-stroke engine, and in the following year Herr Daimler brought out the first practical internal combustion designed to run on liquid fuels. Daimler's engine was almost identical with Otto's later engines, but it also included a device—a carburettor—for vaporizing the fuel and for feeding it, mixed with air in the requisite proportions, to the cylinder.



A Benz Motor-car of the Year 1885

And so came the motor-car. Compared with the present-day luxurious cars, the early models were almost grotesque in their crudity. Nevertheless, the germ of the idea had arrived, and rapid progress began to be made, progress which has been maintained up to the present time.

Otto died at Cologne on 26th January, 1891. Little notice was taken of his decease, and, indeed, even at the present day, he is almost forgotten. Nevertheless, he has left his own memorial in the world, a memorial which you, good reader, will be able to see for yourself if you will step outside your front door and witness the array of motor-cars and similarly propelled vehicles passing by.

FITTING AND ERECTING

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XIII

THE ERECTION OF ENGINES

(contd.)

Locomotive Work. All the sections of this work are prepared in different shops or departments, with the aid of templets and gauges. The erection is done under the supervision of a chargeman. In all engines, however diverse are their types, the first stage is the erection of the frames, level and square over a pit. On these, the engine is built up. The frames will have had their outlines slotted, various horn blocks and angles rivetted on, and holes drilled for various attachments. They are laid on trestles to have centre lines for cylinders and axles marked, and are then dropped into recesses on jacks of the bottle-nose type over the pit, and set at their correct distances apart with stretchers clamped between them. The top edges of the frames are now levelled with a spirit-level and parallel straight-edge, adjustments being made with the jacks near each end and at the centre. The square relations of the frames are tested by cross trammelling or measuring from a rod diagonally to opposite corners, which should show alike. The lateral truth of the frame plates is checked with a straining line set away a little distance from the sides, between which and the wire gauge measurements are taken.

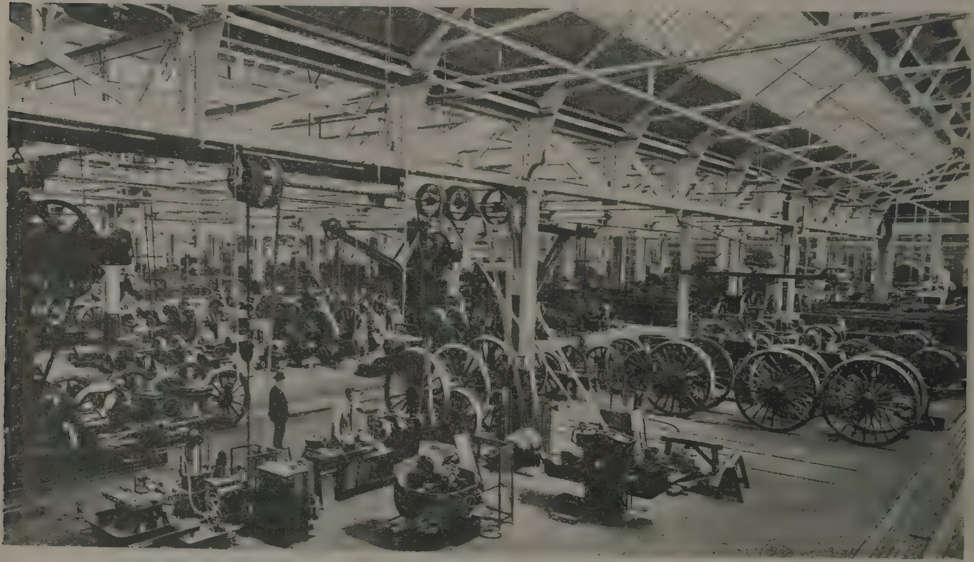


Fig. 79. Wheel Shop, Great Western Railway Works, Swindon

All being levelled square and straight, the frame is ready for the erection of the parts that are contained within it, all of which are fixed at first with temporary bolts and drifts until corrections and adjustments permit of making the permanent fastenings. The heavy casting of the drag plate is set between the hinder portions of the frame and secured with temporary bolts, and also the cross-stay. The horn blocks are fitted into the horns, following which the cylinders and guide bars are inserted. Although these have been tooled to templets and gauges, they are subjected to very careful checking during their erection. Methods vary when cylinders are inside or outside the frames, and whether they are horizontal or inclined. In any case, it is essential that they and the guide bars shall be lined up to the driving axle.

The locations of the parts begin with the cylinders. They are lowered into approximate positions, resting on temporary angle irons clamped to the frames. The height of the cylinders in the frame, and the angle of their inclination when inclined are set from templets bridging the frames, one at the rear of the cylinders, the other to correspond with the driving axle. The top edges correspond with the centres of cylinders and axle. Straining lines are passed from one to the other, tightened with screws, and the cylinders are set concentrically to these lines with a gauge, following which they are bolted up temporarily. The distance from the front of the cylinders to the centre of the driving axle is measured with a rod.

The setting of the motion plate follows. Its distance from the cylinders is set with a templet extending from the lugs on the cylinder cover that receive the slide bar, to the brackets on the motion plate to which the other end of the bar is attached. The brackets are levelled in the transverse direction. To set the bars, a templet is more reliable than working from a strained wire. The bottom bars are set first, gauging their height with a templet passed between their faces and the wire, using thin packings if found necessary. The top guides may be set with a spacing templet. But it is safer to use a templet that gives the areas of the working surfaces of both slide blocks with direct relation to the cylinder. The front cover of the cylinder being removed, a templet cover is substituted, having a large hole through which a light tube is passed extending to the termination of the guide bars. A templet corresponding in thickness and size with the slide blocks is moved along this between the bars, by which they are corrected. Local contacts may be tested with a straightedge and red lead. After all corrections have been made, bolt and rivet holes are reamered flush, and the permanent fastenings made.

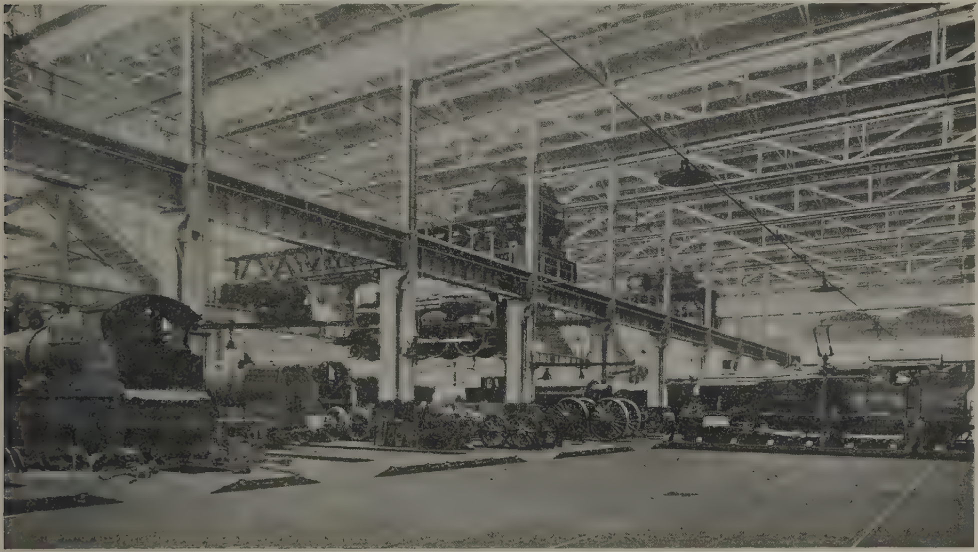


FIG. 80. ERECTING SHOP, GREAT WESTERN RAILWAY WORKS, SWINDON

The setting of the valves and their gear is the next stage. These are laid in the steam chests, and set, first at one end and then at the other in the positions of openings for lead, the spaces being measured with a feeler gauge inserted between the edge of the valve and that of the port. A trammel mark is made on the valve spindle outside the chest, to correspond with each position for reference when the valve is covered in. The cylinder clearance at each end is also marked on the guide bars. The steam chest covers are next attached. The connecting rods are set round the crosshead gudgeons, leaving the big-ends without their straps, freely supported on wood blocking.

At this stage the boiler is mounted on the frames. Correction is seldom required here. The height of the centre of the boiler is checked. It is secured at the smoke-box end with bolts, but is free to slide at the fire-box end on the expansion angle irons riveted to its sides. Detailed fittings succeed. Fire-bars are inserted, the ashpan fastened to the foundation ring, the smoke-box riveted up, blast and steam pipes, splashers, and the boiler lagged. The axle boxes, which have been prepared with their axles, wheels, and springs, are now inserted in their places in the horn blocks, and the big-ends of the connecting rods are closed round the crank pins. The driving wheels are supported on a cradle with friction rollers to permit of the rotation of the wheels during the adjustments of the valve gear. The gear is prepared from models to full size. It is tested in place, the precise method depending on the class of motion used. Essentially it is based on obtaining the dead centres for each crank, working from the slide block with arcs trammelled to the face of the driving wheel, reference being coincident with the lead openings of the valve as scribed on the valve rod, and the cylinder clearances as marked on the guide bars. Considerable detailed work follows: the fitting of the coupling rods, injectors, vacuum brakes, cab, etc.

Figs. 79 and 80, and 81 (facing page 1257), are three views taken in the Swindon works of the Great Western Railway. Fig. 79 is the wheel shop. The lathes are on the farther side. Two tool grinders are seen in the foreground. Fig. 80 is a view in the erecting shop. The erecting pits are seen, and engines in various stages of advancement; two without their boilers, three with them but without the smoke-boxes, and one engine completed is slung in an overhead travelling crane. Other engines may be discerned in distant bays. Fig. 81 is another view in the shop of engines approaching completion. Those on the left are undergoing lagging. One is slung in the crane.

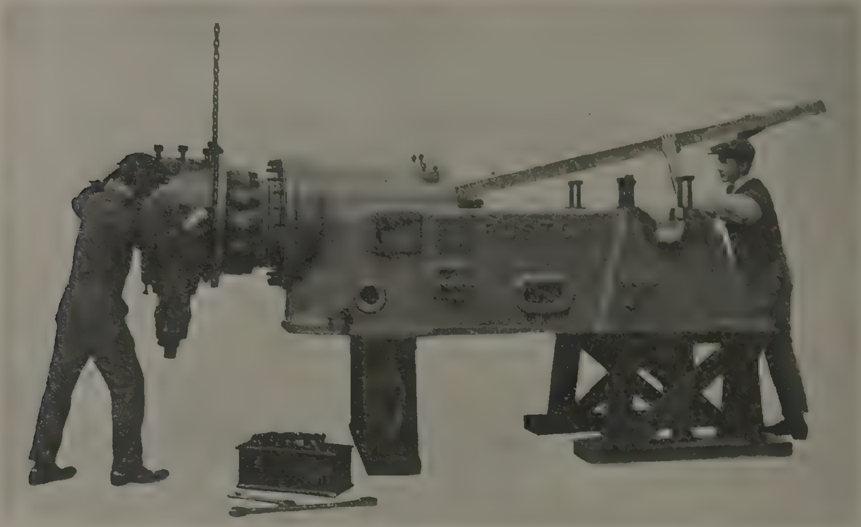


FIG. 82. ASSEMBLING CYLINDER PORTION OF OIL ENGINE

Oil Engines. The erection of these is more simple than that of steam-engines, due to the absence of slide valves and their gears. The valves are lifted with cams. Three stages in the erection of one type of oil engine are shown by Figs. 82–84. Work of this kind built in large numbers is greatly facilitated by the cylindrical character of the principal elements. The bed-plate has the cylinder jacket, cored for water-cooling cast with it. It is bored,

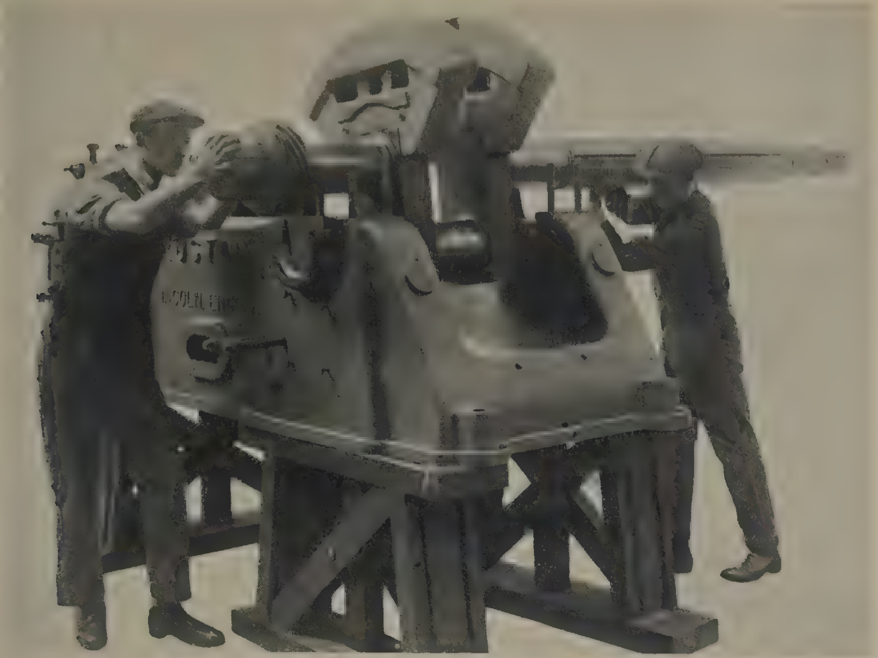


Fig. 83. Putting Crankshaft in Place

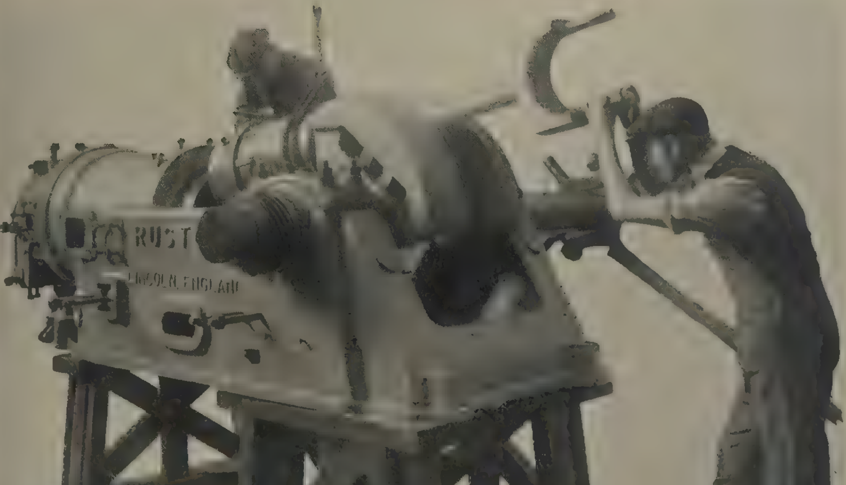


Fig. 84. Inserting Piston

faced, the stud holes at the rear drilled, tapped, and studded, the crankshaft seatings bored, milled, studs and caps fitted, brasses inserted and bored at right angles with the axis of the cylinder. The bed, mounted on trestles, Fig. 82, receives first the cylinder liner, of hard cast iron, easily renewable, and the cylinder head, which contains the valves for air inlet, and exhaust, starting device, and atomizer. the head being bolted up with the studs. The crankshaft, tooled out of a solid steel forging, fitted with its counterbalance weights on the webs, and having the spiral gear that drives the layshaft which actuates the valve gear, the atomizer, and the lubricator is dropped into its bearings, Fig. 83. The trunk piston, Fig. 84, with its spring rings and connecting rod are inserted last. Facings prepared on the side, with studs, receive brackets that carry the bearings for the layshaft, and the mechanism for operating the valves and the governor.

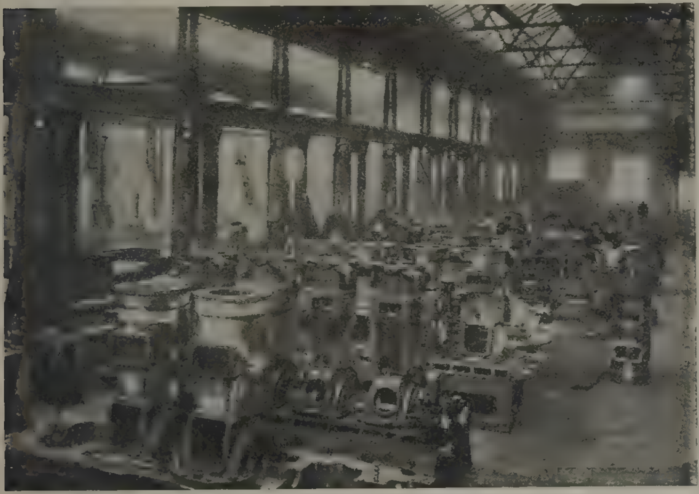


Fig. 85. Erection of Beardmore Crude Oil Engines

Figs. 85 and 86 show a number of the larger component parts of the Beardmore crude oil engine, and a four-cylinder engine completely erected. It is a two-cycle valveless engine, comprising the air chamber or pump in the lower part, into which air is drawn by the upstroke of the piston, and compressed on the downward stroke. At the end of the down stroke ports are uncovered, for the exhaust outlet from the power cylinder, and for the air inlet to it. Through the latter the compressed air enters, and the piston ascending compresses it to a higher degree, and on reaching the top of its stroke the oil, sprayed through an atomizer into the combustion chamber is ignited by the heated air, and drives the piston downwards. Every downstroke is thus a driving stroke.

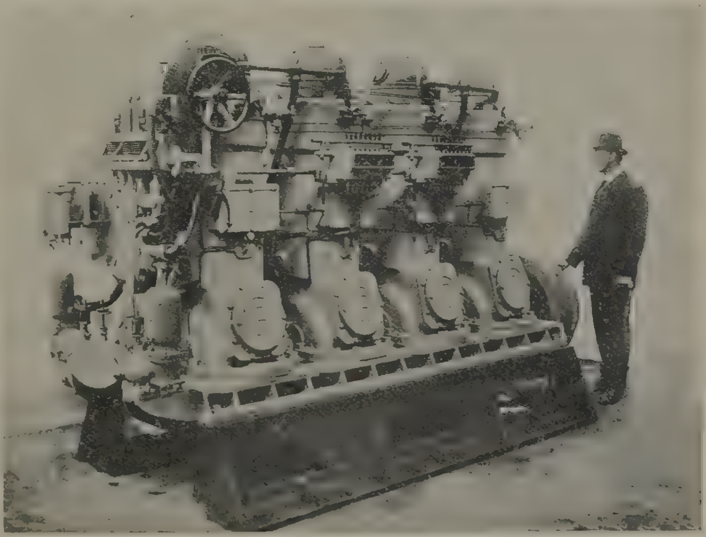


Fig. 86. Beardmore Engine Complete

From one to six cylinders are mounted on a single bed-plate. Main bearings are bored and faced at one setting, and the bushes are lined with white metal. The cylinders and pistons are of close-grained iron. The cylinder cover is water-cooled, and is jointed to the cylinder metal to metal. A water circulating pump is fitted. The stroke of the fuel pumps is varied with a centrifugal governor. All bearings are lubricated with a system of forced feed. Each cylinder is provided with its own separate fuel pump operated by an eccentric working in phase with the crank of the corresponding cylinder. The engine operates equally well in either direction, since the pump driving eccentrics are in phase with the cranks, which renders them suitable for marine service. In engines of this class, though the parts are very numerous, yet the work is as much that of assembling as of fitting. Main parts are fitted with close joints, bored and faced, easily checked with direct methods, and that of the smaller parts is more tedious than troublesome.

(CONCLUSION)

REFRIGERATING MACHINERY

By G. W. DANIELS, M.ENG., WH.EX., A.M.I.MECH.E.

LESSON IV

ICE MAKING

WE will now describe the application of refrigeration to the manufacture of ice.

Two methods are in general use. The first is to place the water to be frozen into cans which are then put in a tank of cold brine until the water is frozen into a block of ice. The second method is to have a tank full of water to be frozen, and a series of cold pipes immersed in the water at regular intervals. The water is frozen round the pipes, the ice block growing larger and larger as the freezing is continued. The ice forming round adjacent pipes ultimately meets and freezes together, the final result being to produce a large slab of ice of practically any desired size, according to the time allowed for freezing. The first method is known as the can system, because the water is frozen in cans. The second system is known as a direct expansion system, because the water is frozen directly round the pipes without the use of any cans or cold brine. These, briefly, are the methods employed, although naturally in an actual plant there are many pieces of auxiliary apparatus.

Fig. 25 shows a typical small ice-making plant of the can type. The ice tank is made of mild steel plates riveted together, and is usually arranged with an internal division running the length of the tank near to one side. The tank is thus divided into two compartments, into the smaller of which are placed the cooling coils. In these coils the refrigerant, ammonia or CO_2, evaporates and cools the brine with which the whole tank is filled. The cold brine is circulated round the tank and over the coils. The cans containing the fresh water to be frozen are placed in the larger compartment of the tank. The cans are made of thin sheet steel, riveted or soldered along the seams, and having a stiff iron band fitted round the top by means of which the cans may be lifted out of the tank when the water is frozen. The ice can is larger at the top than at the bottom in both directions, so that the ice block easily comes out of the can once it has been loosened. Loosening is effected by standing the can containing the ice block in a tank of warm water for a few minutes. This thaws the ice block free from the sides of the can, and the tank used for this purpose is consequently called the "thawing tank."

After removing the ice block, the can is refilled with fresh water and replaced in the freezing tank.

The freezing tank is cased all round with "insulation" to prevent loss of cold to the surrounding atmosphere. This insulation may consist of a hollow timber casing packed with granulated cork, or often consists of compressed cork slabs built round the tank with bitumen or cement.

Fig. 26 shows a section through a large ice plant. In this the ice tank is placed on an upper floor, the lower story serving as the ice store. In such large plants the ice cans are fixed in frames so that a whole row of cans may be lifted at once, a crane being used for this purpose. The crane and thawing tank will be seen on Fig. 26, also the tipping cradle, which is a device for turning the row of cans on to their sides to shoot the ice blocks from them. The crane then lifts the cans up again and places them under the filling tank from which they are refilled with fresh water. In some cases the water is cooled before being run into the filling tank.

Fig. 26 also shows the ice crusher, placed level with the floor of the ice tank room so that ice blocks can be easily dropped into it, crushed, and delivered on to the floor beneath. The lift for raising ice blocks from the store to the crushing floor will be noticed, and also the shute down which ice blocks can be directly slid into the ice store. In this large tank four 18 in. diameter propellers circulate the brine over the cans. The bottom of the ice tank comes directly into the ice store and serves to keep the ice store cold. The whole arrangement forms a very good typical layout for a can ice plant of large size.

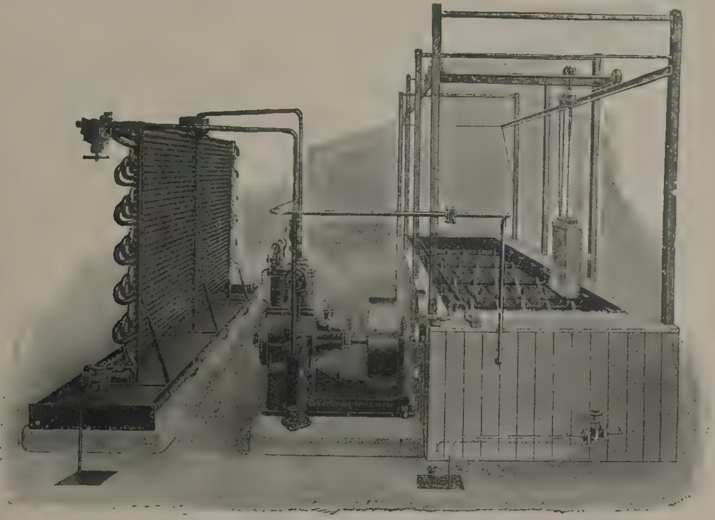


Fig. 25. Small Ice-making Plant
(The Haslam Foundry and Engineering Co., Ltd.)

Clear and Opaque Ice. A block of ice frozen from still water in a can would come out of a white colour and opaque. Such ice is not so hard, lasting, nor valuable as a clear crystal block of ice. The opaqueness is due to air and gases originally present in the water to be frozen. These gases are liberated at the moment of freezing in the form of small bubbles, and in still water these bubbles are imprisoned in the ice block, causing the opaqueness. To overcome this the water in the can should be kept stirred up whilst freezing, and this can be conveniently done by suspending a small brass pipe in the centre of each can down which clean cold compressed air is blown. The large bubbles of air rising from the end of this tube keep the water in motion, with the result that the minute bubbles forming the whiteness are washed away.

In some cases air pipes are permanently connected to the cans, the air issuing from a small nozzle in the bottom of the can. Two systems of air agitation are in use; the high pressure, using air at about 18 to 20 lb. pressure, and the low pressure, using air at 2 to 3 lb. pressure.

Ice Crushers. In many cases the ice is required not in the form of blocks but crushed into small pieces. Fig. 27 shows a machine for this purpose. It consists essentially of revolving rollers carrying strong teeth and contained in a housing. The ice block is put into the hopper on top of the machine and is rapidly picked into pieces by the teeth on the rollers. The crushed ice leaves underneath.

Time for Freezing Ice. The following figures give some idea of the time required to produce

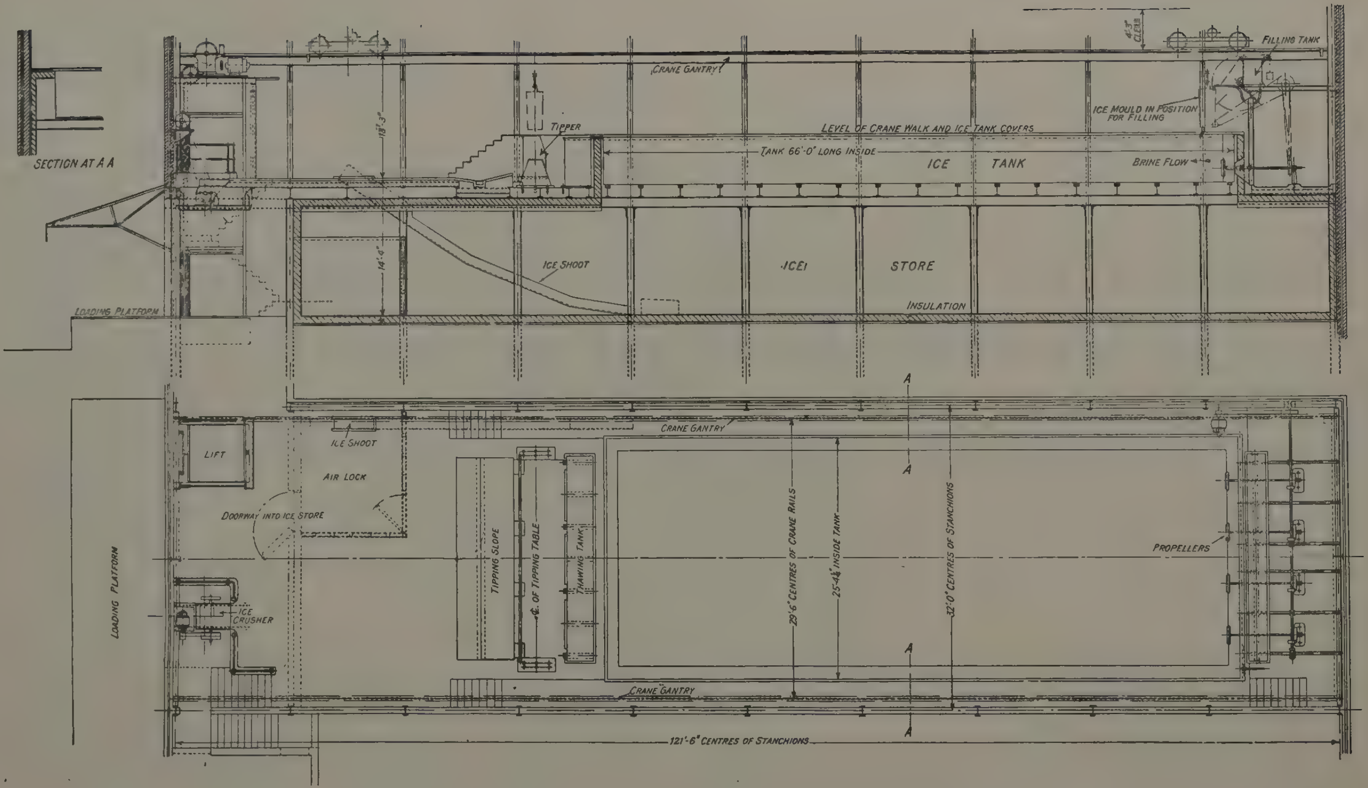


Fig. 26. Sectional Elevation and Plan of Large Ice Plant

can ice of various thicknesses, assuming the temperature of the cold brine in the freezing tank to be 15 to 16° F.—

Thickness	4 in.	5 in.	6 in.	8 in.	10 in.
Hours to freeze	7	11	16	28	44

With brine at a lower temperature, the time for freezing would be less of course.

To freeze 1 lb. of water at 32° F. into ice requires about 144 B.Th.U.'s to be removed from it, but to make 1 lb. of ice in commerce requires much more heat than this to be removed, and in practice we have to allow for the refrigerating machine removing 240 to 300 B.Th.U.'s for every pound of ice made. This additional amount of refrigeration is necessitated by the cooling of the water down to freezing point, by the subsequent cooling of the ice below freezing point, and by the leakages of heat into the cold parts of the plant.

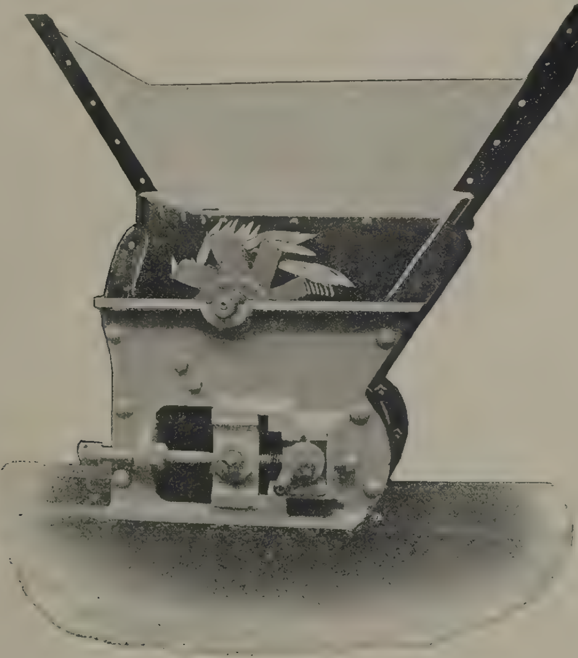


FIG. 27. ICE CRUSHER

For every ton of ice made per 24 hours it is usual to allow about 270 ft. of $1\frac{1}{4}$ in. bore piping in the ice tank, or an equivalent surface in other size pipe.

The size of the ice blocks produced in a can plant will vary from a few pounds to, say, 3 cwt., according to the size of the plant, while in direct expansion plants the size of the block produced may reach half a ton or more. Such large pieces are sawn up into smaller ones of a convenient size and weight for marketing.

The brine used in refrigerating plants is always made from calcium chloride and not sodium chloride or common salt. This is because brine made from common salt has been found to be corrosive to the iron coils, and, further, common salt brine is liable to freeze at the temperatures often met with in refrigeration.

OUTLINES OF AUTOMOBILE ENGINEERING

By J. L. Milligan, B.Sc., (Vict.), M.I.A.E.

LESSON VIII

CLUTCHES AND GEARBOXES

The usual petrol engine has a very uneven turning moment even with six cylinders and is incapable, in its modern form, of self-starting, as opposed to hand or electric starting. It is therefore necessary to have some method of releasing the engine from the transmission so that it may be run idle when the car is at rest.

As it is necessary also to bring the car into motion without stopping the engine, a clutch which allows slip between two members until they arrive at a common speed is required.

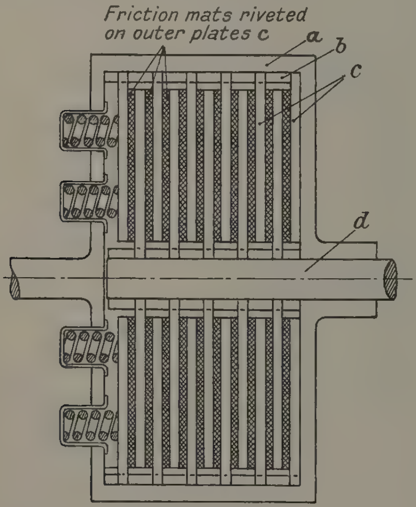


Fig. 35. Diagrammatic View of Multi-plate Clutch

The simplest type is where two discs attached to the engine shaft and transmission shaft respectively are pressed together until they transmit the whole torque without slip. This is an unpractical scheme owing to the high pressure required.

In order to reduce end-pressure the cone clutch is used. This is in several forms differing only in detail. The angle of the cone should be not less than the " angle of friction " of the materials, or it will " stick " in and require more force to release it than is necessary to hold it in.

Clutches are always held " in," i.e. in driving contact, by a spring, and " out " by pressure on a lever under the left foot, called the clutch pedal.

The inverted cone is the simpler form of cone clutch and gives excellent service, although there is not much to choose between the types.

It is not so easy to adjust the spring pressure on the inverted as on many external cones. This is a blessing in disguise. Cone clutches are gradually disappearing in favour of the single disc or multiple disc.

At one time cones of hardened steel working in cast iron or gun metal and well lubricated were common, but their lubrication is of great importance, and the public will not look after things under floor boards.

As the fly-wheel has no mechanical function to perform, except carrying the engine over the " dead points " where no work is being done by the pistons, it is usually made to act as part of the clutch.

In order to make a quick start it is possible to run the engine very quickly and let the clutch in so that part of the fly-wheel momentum is transformed into car momentum at a lower engine speed. (Note it is *not* energy transformed as some is lost in heat. This is a common error.) As there is no room nor necessity for such a practice, fly-wheels should be kept as light as is consistent with smooth running when at rest or driving slowly in traffic. If there is any necessity to speed up an engine before allowing the clutch to engage in order to make a start there is something wrong with either engine, clutch, or driver.

Plate Clutches. The plate clutch is the commonest type of clutch to-day. It is of several forms as single or double plate. A diagrammatic view of a multiple plate clutch is shown in Fig. 35.

A drum *a* attached to the fly-wheel and having keys in its inside at *b* has plates fitted in it which can slide along axially, but cannot rotate because of the keys.

These plates have holes in the centre large enough to clear the keys on a shaft *d* which transmits power to the driving gear.

Plates keyed on *d*, but sliding along it, are interposed between the plates *c*. If pressure is put on the outside plates (there should be an odd number of "outers") by springs as shown, then each "inner" is pressed on both sides by outers, and whatever torque is necessary to slip one inner over one outer, under that spring pressure, is multiplied by the number of surfaces which can slip. The actual surface in contact does not matter in the least, as far as grip goes, but when too small a rubbing surface is used the heat generated cannot get away quickly enough and the temperature of the rubbing surfaces rises. The majority of materials used for clutch linings have a coefficient of friction which falls as the temperature rises. This is especially true of leather. If therefore the temperature once rises, the clutch will slip and generate more heat until it burns and becomes useless.

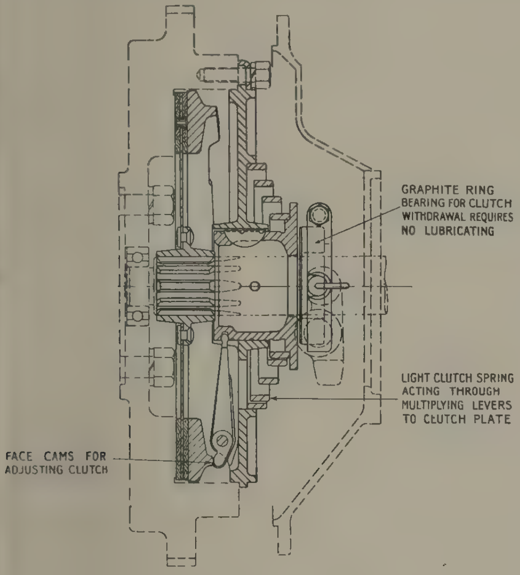

GRAPHITE RING BEARING FOR CLUTCH WITHDRAWAL REQUIRES NO LUBRICATING
LIGHT CLUTCH SPRING ACTING THROUGH MULTIPLYING LEVERS TO CLUTCH PLATE
FACE CAMS FOR ADJUSTING CLUTCH

Fig. 36. Borg-Beck Clutch
(Automotive Products Co.)

A few calculations on the magnitude of forces dealt with can be considered here. Assume the case of an engine developing 25 b.h.p. at 1,000 r.p.m. The average torque on the fly-wheel is 25 × 63 lb.[1] at 1 in. radius. Assume mean radius of clutch rubbing surface is 7 in. The tangential force is then $(25 \times 63) \div 7 = 225$ lb. If the coefficient of friction of the liner on the metal face is ·2 (some linings are higher than this), the end pressure necessary just to stop slip is $\frac{225}{0{\cdot}2} = 1{,}125$ lb.

If a rubbing face is provided on both sides, half this amount is necessary.

In a plate clutch containing 10 inners and 11 outers, there are 20 rubbing faces and the pressure will be $\frac{1125}{20} = 56{\cdot}2$ lb.

The great difficulty in using such a type of clutch is its length, and in order to allow for inequalities in thickness of plates there must be a definite separation of the surfaces. The more plates used, the more travel is necessary on the end plate to free the clutch—and as liners wear, the clutch requires adjustment.

The present practice is to use only one or two plates, and use levers to provide the pressure necessary for disengagement without heavy pressure on the sliding collar. It is essential to do this, as if heavy pressures are used in withdrawal the crankshaft is thrust backward or forward, and there must be proper provision for this on the rear bearing in order to prevent "float" occurring. Fig. 36 shows a well-known plate clutch.

Cone Clutches—Angle of Cone. If the cone runs on an extension of the fly-wheel shaft, it can only travel in direction of arrow *T*, Fig. 37. Pressure between faces at a point *P* can vary from the normal to an angle equal to the angle of friction, i.e. *PA* and *PB*. The angle of friction ϕ is tan μ.

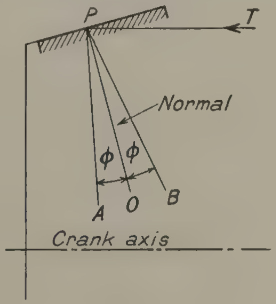


Fig. 37

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[1] Every student should memorize this figure 63 which is the torque in in.-lb. producing 1 h.p. at 1,000 r.p.m.

In calculating the necessary pressures it is not sufficient to take cognizance only of the angle of the *cone* to find the required amount, the angle of friction is to be accounted for as well.

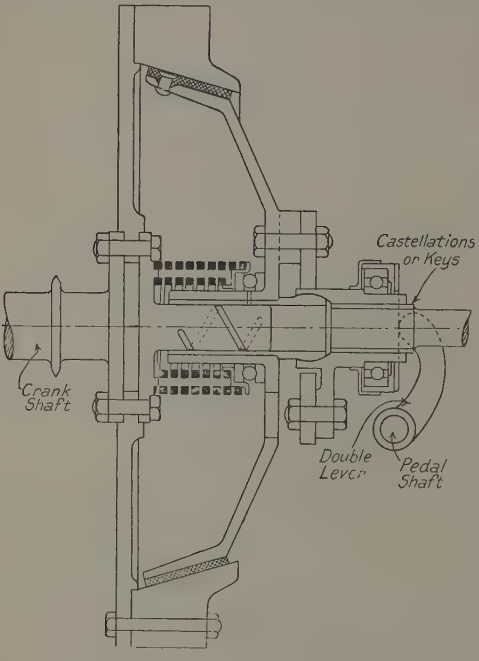


FIG. 38. INVERTED CONE CLUTCH

If the angle of the cone is less than the angle of friction then the clutch may continue to function if pressure is removed, and force will be required beyond the spring pressure; on the other hand, pressure beyond the ideal spring pressure is necessary to bring it tightly into contact.

The "fierceness" of a clutch is due to the angle being too small for the material used. Many cone clutch fabrics contain dressings which soften and exude at high temperatures so that after much "slipping," either due to crawling in traffic or misadjustment, they will stick in.

For small cars and motor-cycles cork studs are in use as friction material, running in oil and are an excellent material, but are very quickly destroyed if heating occurs. They will stand considerable abuse.

The parts of a clutch which only revolve when in engagement, and not when the engine is running with clutch disengaged, must be as light as possible and small in diameter in order to reduce momentum to the lowest amount. This is necessary for gear changing.

Many types of clutch have been tried, but the two types, cone and single or double plates, are now almost exclusive. Hydraulic clutches in which oil was pumped through adjustable restrictions and electro-magnetic clutches have both been tried and discarded for automobile work, also coil clutches, which are excellent on mill machinery where the torque is always in one direction, whereas in descending hills on a car the car drives the engine and releases the coil clutch.

Gearboxes. If the curves of horse-power to engine revolutions per minute be plotted, they will be found to have a shape as shown in Fig. 39 for a modern high efficiency engine and for a low compression, low speed engine respectively. Suppose we plot in Fig. 40 the horse-power necessary to run at varying road speeds on level, up hills of differing grades or on slight down grades. This is a function of car weight, wind resistance (form of vehicle), road surface, and transmission efficiency from engine to road wheels.

These are not constant for any given vehicle. Weight is obviously variable. Wind is of great importance at high speeds and whether with or against the car's motions.

Road Resistance varies with surface and foundations of road, with class of tyre, and on roads of bitumen or tar surfaces with temperature. A pedal bicycle will give more reliable information on these points. Gradients rideable after rain are not rideable when hot and sticky. At slow speeds soft tyres are at a disadvantage in power required to move them.

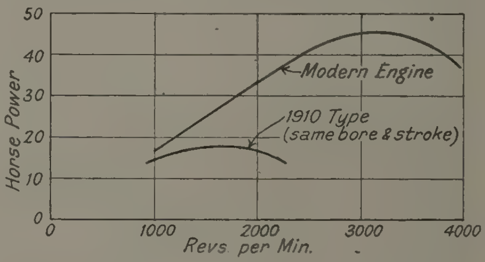


FIG. 39. REVS. PER MINUTE—HORSE-POWER CURVES

It is obvious that if an engine with a curve as

Fig. 39 is required for a car whose road requirements are as Fig. 40, it is important to relate the engine speed to the car speed at more than one point. If the speed reduction to the wheels is made so that the maximum "peak" of the power curve is fitted to the 1 in 10 gradient curve, as shown, the car will not attain much higher speed on the level than on that gradient.

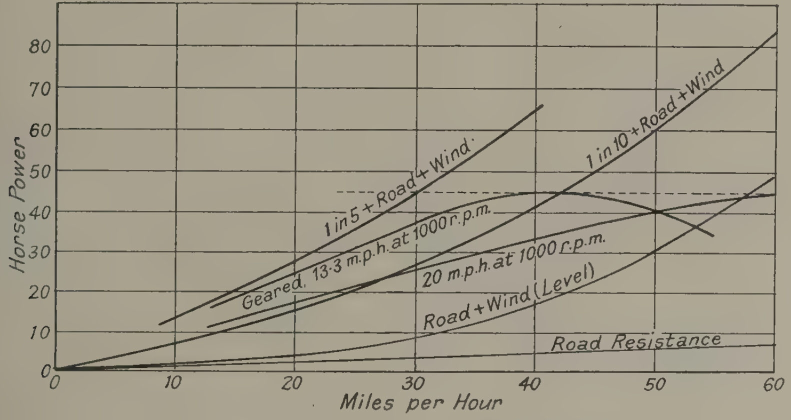


FIG. 40. POWER CURVES FOR CAR ONE TON WEIGHT ENGINE AS FIG. 39

Showing engine "peak" arranged for one in ten grade and for level.
Note. Horse-power for wind resistance varies as cube of speed.

In order to accommodate it to the level conditions, the peak of the power curve must come at another road speed.

If a car had to operate with known loads over a known gradient it would be possible to provide it with a single gear, but in practice a choice of three or four is given to cover all circumstances.

In passing we may note that two of the cheapest vehicles have but two changes.

The relationship of the gears in a gearbox is a matter of compromise.

First consider the three-speed box. In private cars the ratio from top gear to bottom gear is seldom over 3 to 1, that is top 100, low 33. If the second is placed at 66 it is a big step first to second, so if a hill is being climbed well on first it is a risk to bring the engine down to half its speed by changing up and maintaining the same road speed.

If, on the other hand, we bring it down to 50, the gap occurs between second and third and, if for a delay on a slight grade, the driver has to use second he has difficulty in regaining his road speed on changing to top.

The second is always a compromise in this case. Some purists calculate it as the geometric mean of the gear ratios, holding that the changing will then occur at the same engine speeds and the driver will know by the note of the engine when to change. In these days of silent six-cylinder engines and closed cars the driver does not know the "note" of his engine.

The usual settlement is by the designer seeing how the job lays out for bearings, and also to see that as far as possible the same tooth numbers shall be used in the box twice to cheapen production. This is the *real* reason for the geometric mean box.

In four-speed boxes a lower first is usually provided than in three-speed boxes, but not so low as to make the second correspond to the old third.

There are many types. Some keep a close ratio top to third for fast climbing on usual main road hills. The double reduction in the box causes lost power, and hence if third is too near top the extra loss will give no advantage in changing down to third.

In such cases changes must be made at high speeds.

The 100, 75, 50, and 25 per cent box is common and is an excellent box, but second is high for starting from, and first too low.

The design is again dependent on the choice of repetition of teeth, especially when ground gears are used in order to cheapen production. In the next lesson we will consider some of the practical points in gearbox design, and gear changing.

POWER TRANSMISSION BY BELTING

By W. G. Dunkley, B.Sc.

LESSON XI

BELT SHIFTING FORKS, BELT SHIFTING ARRANGEMENTS, QUARTER TWIST DRIVE, GUIDE PULLEY DRIVE

Belt Shifting Forks. In the case of belt driven machines it is usually necessary to have some means of stopping the machine. Some form of clutch may be provided in which case the belt will run continuously so long as the mainshaft is running from which it is driven. Fast and loose pulleys may be provided in place of the clutch and, in this case, the belt is moved from the fast pulley which is keyed to the shaft on to a pulley which runs loose on the shaft. Fig. 9 shows a common form of belt shifting gear. The belt runs between two forks as shown. These forks are carried on a bar *A*, which is supported loosely in slots *B*, in the hangers. Suitable means is provided for moving the belt fork bar along so that the belt forks constrain the belt to move along from one pulley to the other. In Fig. 9, the bar is a rough black bar, and is not machined. The forks are generally of malleable cast iron. The diameter of the forks may be 1 to 1¼ in. for general cases. It will be seen that as the belt is being moved by the forks the edge of the belt will be rubbing against the fork. This is not good for the belt and so, sometimes, the forks are made as in Fig. 10. In this case we have two common steel rods *A*, on which are placed lengths of suitable standard pipe. The pipes are kept in place, and the two forks braced together by the connecting plate *C*. As the belt is moved the pipe can rotate and so the friction between the belt and fork is considerably reduced.

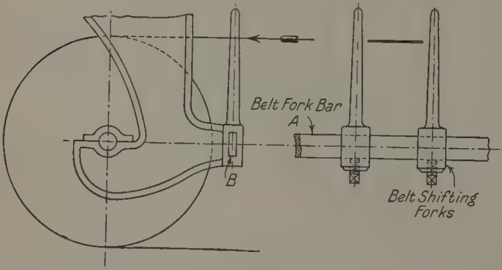


Fig. 9

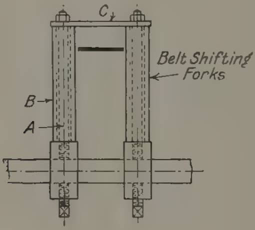


Fig. 10

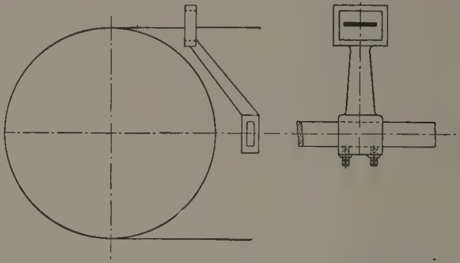


Fig. 11

Position of Belt Shifting Forks. It is very important that the forks should be placed so that the belt is struck on the entering side, and not on the leaving side.

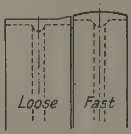


Fig. 12

Fork for Large Pulleys. It is very desirable that the fork should strike the belt as near the pulley as possible. When the pulley is of fairly large diameter, if the forks are made straight, the distance between the fork and the pulley leaves a somewhat flexible length of belt between the fork and the pulley surface. In order to reduce this length the forks are made with a drop, as shown in Fig. 11, and of a stronger construction as will be seen.

Fast and Loose Pulleys. In order to relieve the tension on the belt, and the load on the bearing, the loose pulley is sometimes made smaller in diameter than the fast pulley. The loose pulley tapers up to meet the fast pulley, as shown in Fig. 12. When a pulley is loose the drag of the belt in moving across the face is able to tilt the pulley and thus concentrate the load on the two edges of the *boss*. The boss of the loose pulley should be made as liberal as possible in length.

Reversing Belt Drive. A reverse drive may be provided for a belt driven machine by driving the machine from a countershaft, and arranging a reverse drive to the countershaft. A countershaft is a short shaft driven from the main shaft which usually runs the full length of the shop. The main shaft runs all the time at a given speed, and the countershaft is generally necessary to provide a means of stopping the machine, and to take care of speed requirements at the various machines. Referring to Fig. 13, we have a set of three pulleys *A*, *B*, and *C*, driven by an open belt from one wide pulley on the main shaft and a set of three pulleys *D*, *E*, and *F*, driven by a crossed belt from another wide pulley, also on the main shaft. The pulleys *D*, *E*, and *F*, will, therefore, run in the reverse direction to the pulleys *A*, *B*, and *C*. Now when the open belt is on pulley *A*, the crossed belt is on pulley *D*. The countershaft is then driven in the direction indicated on pulley *A*. The two belts are moved simultaneously across the pulleys, so that when the open belt is on *B* the crossed belt is on *E*. Both these pulleys are loose on the shaft and, therefore, the countershaft is idle. As the two belts are moved farther along the open belt moves to the loose pulley *C*, and the crossed belt to the fast pulley *F*, and the countershaft now runs in the reverse direction as indicated.

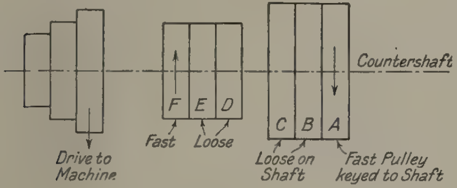


FIG. 13

Reversing Belt Drive for Planing Machine. The three pulley design shown in Fig. 13 occupies too much floor space to be really suitable for a planer drive. A reversing drive for a planing machine is illustrated in Fig. 14.

The table of the planing machine carries two dogs or stops in a tee groove along the side. These dogs can be adjusted along the table to effect the reverse at the end of the required length of stroke. At the suitable part of the stroke the dog pushes over a tumbler lever. The angular movement of this tumbler lever is used to transmit motion, by means of a system of shafts and connecting rods, to the cam bar, shown in Fig. 14. The strap forks *B* and *C*, are double levers pivoted on studs *K*, carried by a bridge bracket *L*. The short arms of the

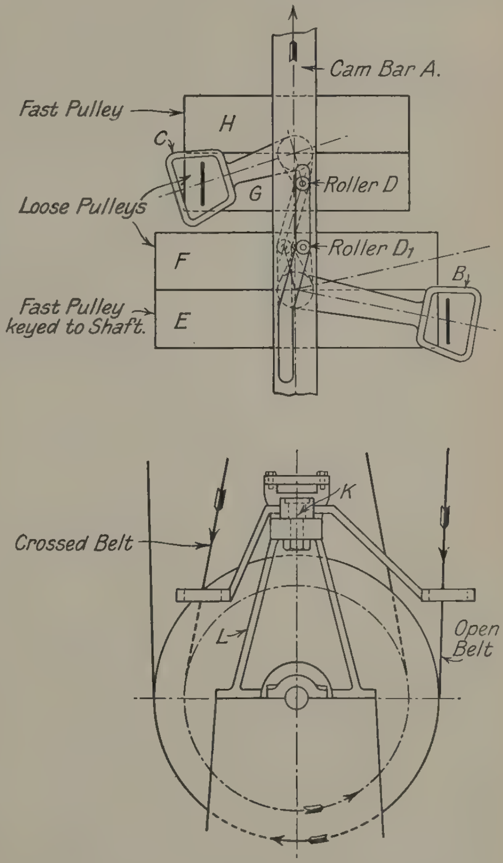


FIG. 14

strap forks carry hardened steel rollers D and D_1 which slide in a cam slot in the cam bar as indicated. The open belt should run on the large diameter pulleys and the crossed belt on the smaller pulleys, as this gives a larger angle of lap on the smaller pulley than if the open belt were on the small pulley. In the position shown, the open belt is driving on the fast pulley E, and the crossed belt on the loose pulley G.

Now as the cam bar is moved in the direction indicated by the arrow, the inclined part of the cam slot engages on the roller D_1 and causes the strap fork B to rotate on the stud and thus to move the belt on to the loose pulley F. At the same time, the straight part of the slot engages the roller D, and no motion is given to the fork C. The two belts are now on the loose pulleys. As the motion of the cam bar continues the inclined part of the slot engages roller D and throws the belt from G on to the fast pulley H. At this time the roller D_1 rides in the straight part of the slot. The crossed belt is now driving the machine in the reverse direction, the open belt being on the loose pulley. The operator can actuate the tumbler lever by means of a hand lever, and so can reverse the machine by hand at any moment, or he can throw the cam bar into the position in which both belts are on the loose pulleys and so stop his machine.

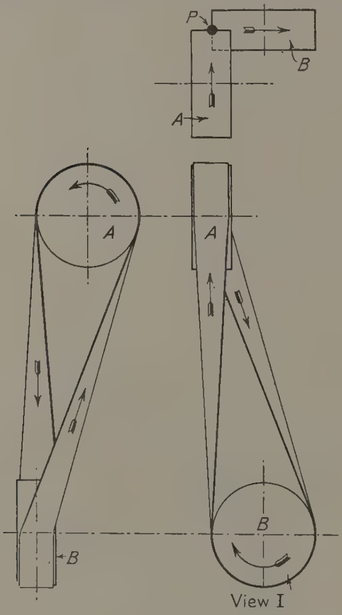


FIG. 15. QUARTER TWIST DRIVE

Quarter Twist Belt Drive. A quarter twist drive is illustrated in Fig. 15, in which the shafts are at right angles. In these drives the pulleys must be arranged so that the belt *enters each* pulley in a path at right angles to the axis of the pulley. Consider view (1). If we move B at right angles to the paper we should not disturb the plane of entry of the belt on to pulley A, but if the belt is to *enter* pulley B in a direction at right angles to its axis the only possible position for pulley B is as shown in the plan view. If the pulleys are to be arranged at an angle other than 90 degrees, the pulleys may be pivoted about the point P.

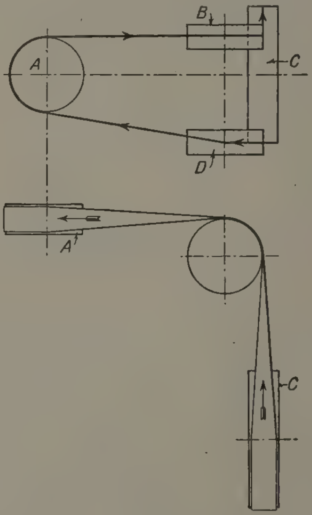


FIG. 16. GUIDE PULLEY DRIVE

Guide Pulley Drive (Fig. 16). To keep to the principle stated for quarter twist drives, the following conditions are necessary. The plane through the middle of B must be tangent to the face of pulley A. The plane through the middle of C must be tangent to the face of pulley B. The plane through the middle of D must be tangent to the face of pulley C. The plane through the middle of A must be tangent to pulley D. Pulleys B and D are not necessarily on the same centre. Pulley B may be raised vertically and D moved horizontally without disturbing the above conditions.

(CONCLUSION)

OUTLINES OF LOCOMOTIVE ENGINEERING

By A. MORTON BELL, O.B.E., M.I.MECH.E., M.I.LOCO.E.

LESSON VI

THE ENGINE

THE engine comprises the mechanism whereby the energy developed by the combustion of fuel in the fire-box of the boiler is converted into motion through the agency of the steam generated. It embraces the cylinders, pistons, connecting rods, and cranks, with the necessary "motion" or valve gear, to effect the correct and most satisfactory distribution of the steam, and direct the movement of the engine either backwards or forwards.

The cylinders may be two, three, four, or even more in number, placed either inside or outside of the framing, or both. Whilst locomotives were of moderate dimensions and power, two cylinders usually sufficed, and the favourite location in this country was inside the framing. As the size and power of engines developed and the restrictions imposed by the gauge of railway, with the consequent width apart of the members forming the framing, preventing them from being placed there, they had to be arranged outside, and now most of the larger engines running are so designed. Limitations in "overall" width have, however, again dictated further re-arrangement, as cylinders beyond a certain diameter cannot pass loading gauge restrictions (see Fig. 1, on page 862). Consequently, recourse has had to be made to employing three and four cylinders, the additional ones being placed usually between the framing, or vice versa.

In the case of compound engines, a combined arrangement of cylinders inside and outside is frequently resorted to, although there are cases when the large size of even the smaller high pressure cylinder precludes their location there, as in the huge "Mallett" engines running.

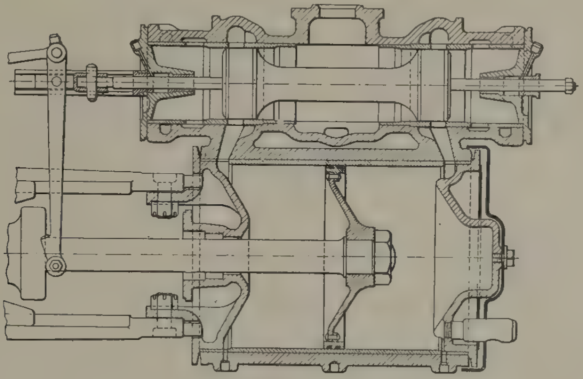


FIG. 19. SECTION OF CYLINDER WITH VALVE CHEST

The cylinders of a locomotive are generally made of a special close-grained, hard, cast iron, carefully manufactured, to ensure good, clean, and perfect castings. Cast steel is now being employed for some of the very large cylinders used on locomotives of exceptional dimensions.

The above diagram, Fig. 19, gives a section through a cylinder and its steam ports. The piston, in this case, is of steel, this material being lighter than cast iron for a given strength. The piston-rod, also of steel, is fitted to the piston with a coned end, secured by a screwed nut and locked.

To secure distribution of the steam admitted to the cylinders from the steam chest, valves are necessary, and these represent the result of much invention, controversy, and experience, and form a most interesting subject for students of locomotive engineering.

The simple "D" slide valve (usually of bronze), of which a sketch is given in Fig. 20, answered all requirements of the early locomotives.

There are three ports, or passages, on the cylinder face; the outer ones communicating with the two ends of the cylinder and either side of the piston working in it, whilst the centre one is for the exhaust and leads to the blast pipe. If the valve is moved to the right or left, it is obvious that steam will be admitted to that end of the cylinder of which the port is uncovered, whilst the central portion, or cavity, of the valve will form a passage for steam escaping from the opposite side.

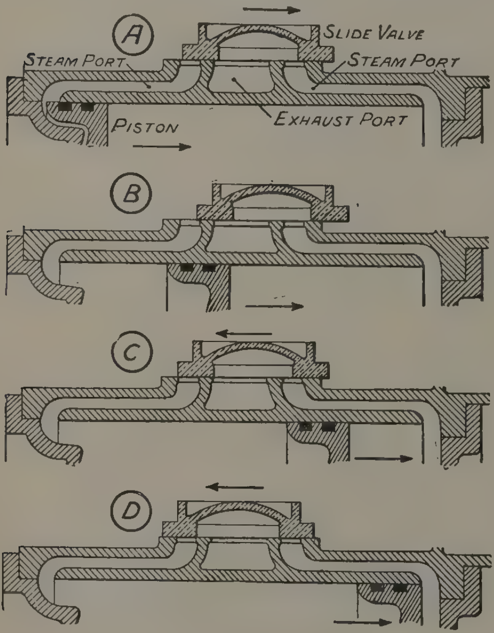


Fig. 20. Action of the "D" Slide Valve

In the drawing, Fig. 20, *D*, the valve is shown in mid-travel, and it will be noted it more than covers the ports on either side; this extension of the face of the valve is termed the "lap," and is provided so that the valve shall cut off access of steam to the ports before the piston in the cylinder reaches the end of its stroke. This latter will then be completed by the expansion of the steam imprisoned in the cylinder. The valve continuing its movement in the direction of the arrow will next cause the port to be opened to exhaust by way of the cavity of the valve, whilst on its other side the edge of the valve will uncover its port to steam.

As the piston immediately approaches the end of its stroke, the valve is arranged to slightly open by an amount called the "lead"; this is secured by setting the eccentrics on the axle (with link motions) in such a position termed the "angular advance," as will cause the valves to be moved the required amount in advance of the movement of the piston.

The operation of the slide valve will be best followed by reference to the diagram, Fig. 20, which gives in *A*, *B*, *C*, and *D* the relative positions of the piston and valve during one-half of a revolution of the crank; it is easy to reverse the movements, and so complete one cycle.

At *A*, the piston is shown at the front of the cylinder about to commence its stroke, and the valve is seen just opening the port by the amount of "lead." At *B*, the piston has moved to nearly half of its stroke, and the valve has opened to the full extent of its travel. At *C*, the piston has reached three-fourths of its stroke, whilst the valve has reversed and has closed the port, due to its "lap." At *D*, the piston is shown as being on the point of finishing its stroke, whilst the valve having closed the port to steam, is commencing its opening to exhaust. At the other end of the cylinder it has just closed the port to exhaust, and steam in the cylinder is being compressed by the piston, forming an elastic cushion against the shock of reversal of movement. Immediately before the piston stops, the valve will be opened by the amount of its lead, as shown in *A*, and the sequence of events is repeated for the other end of the cylinder. These fundamental operations of the slide valve hold good for all other steam distribution devices, although they may vary in the way they actually carry out the procedure.

The adoption of higher steam pressure, superheating, etc., have necessitated improvement on this device, which relied entirely upon flat, sliding surfaces for all movements, entailing much friction. In the original installations of cylindrical valves intended to replace these, trouble was experienced with condensed water getting jammed in the cylinders and unable to escape (with the old slide valves, it lifted them), but all this has now been satisfactorily met by providing flexibility in the packing rings, etc.

A typical and well-designed piston valve is

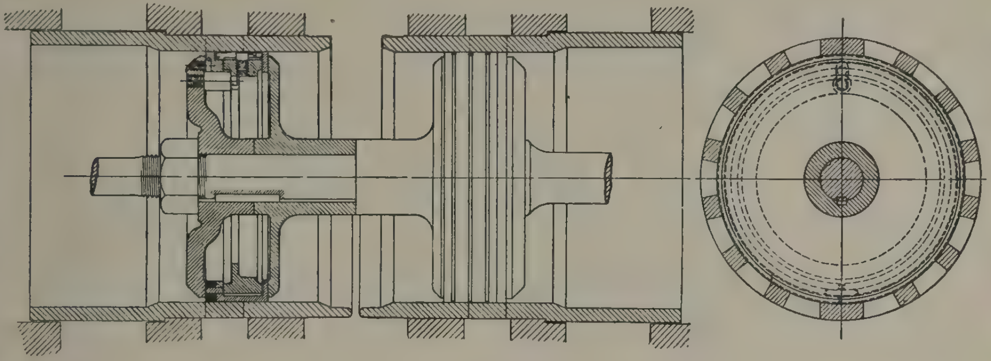


FIG. 21. PISTON VALVE

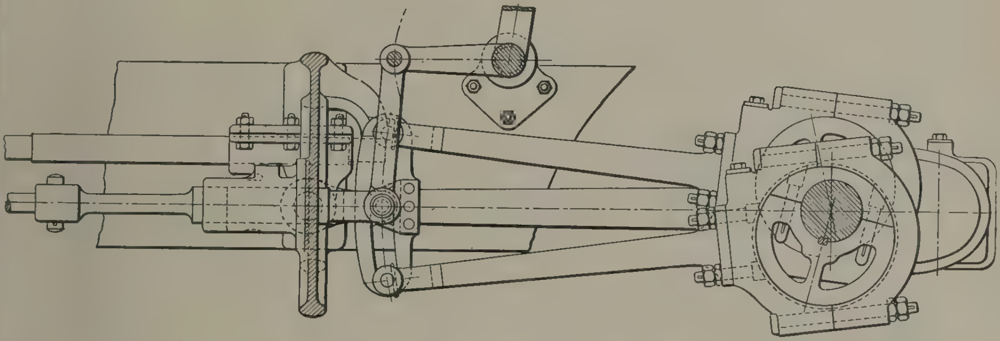


FIG. 22. STEPHENSON'S LINK MOTION

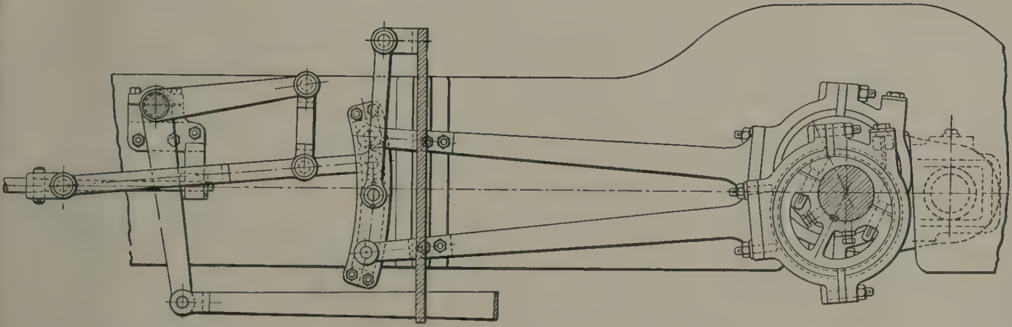


FIG. 23. GOOCH'S STATIONARY LINK MOTION

shown in the drawing, Fig. 21, which is arranged for " inside " admission ; that is, the steam from the boiler is introduced between the pistons of the valve, which is directly opposite procedure to the working condition of an ordinary *D* slide valve. A great advantage of this arrangement is that the valve spindle, " gland," or stuffing-box is only subjected to the pressure of exhaust steam, and not to high pressure, superheated steam, as it would be if conditions were reversed.

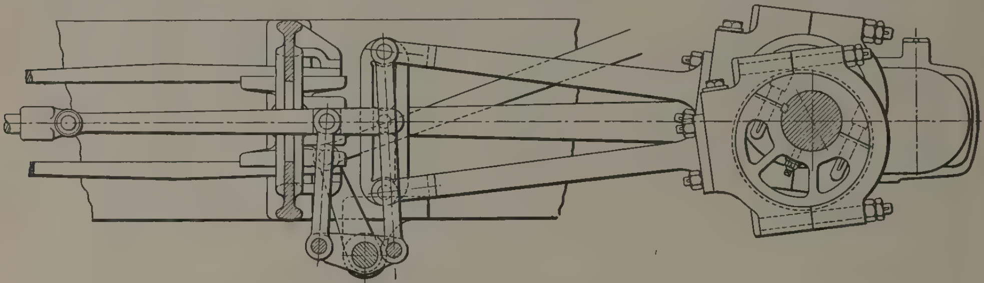


FIG. 24. ALLAN'S STRAIGHT LINK MOTION

Coming now to the motion, or means of operating the valves, to secure the required distribution of steam, there are many designs in use. A description of the more common type will be interesting in the order in which it may be said they were introduced—

1. The Stephenson " shifting " link.
2. The Gooch " stationary," or " box " link.
3. The Allan " straight " link.
4. The Joy's radial gear.
5. The Walschaert's gear.

The Stephenson was undoubtedly the gear most used in this country until recently. It is now being gradually supplanted by the Walschaert, a gear of almost precisely the same age.

In the Stephenson motion, two eccentrics are employed for each cylinder, and the general arrangement will be readily followed from the diagram, Fig. 22. The connecting link is known as the " expansion " link, and it is evident that the block, or die, which slides in it, and is connected to the valve spindle, will receive motion in proportion to its proximity to either the top or bottom of this link: at the centre it will be mid-gear, remaining all but stationary, merely oscillating on its pin. The operation is simple, the driver just lifting or lowering the expansion link as he desires his engine to go, backwards or forwards, or placing it at any intermediate position for controlling the travel of the slide valve, and consequent " admission " or " cut-off " of steam. With the Stephenson motion, the lead of the valve does not remain constant, but varies somewhat in the different positions of cut-off.

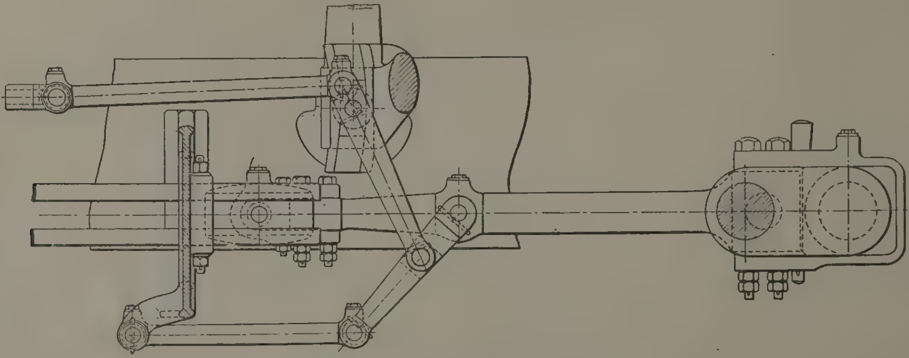


FIG. 25. JOY'S RADIAL GEAR

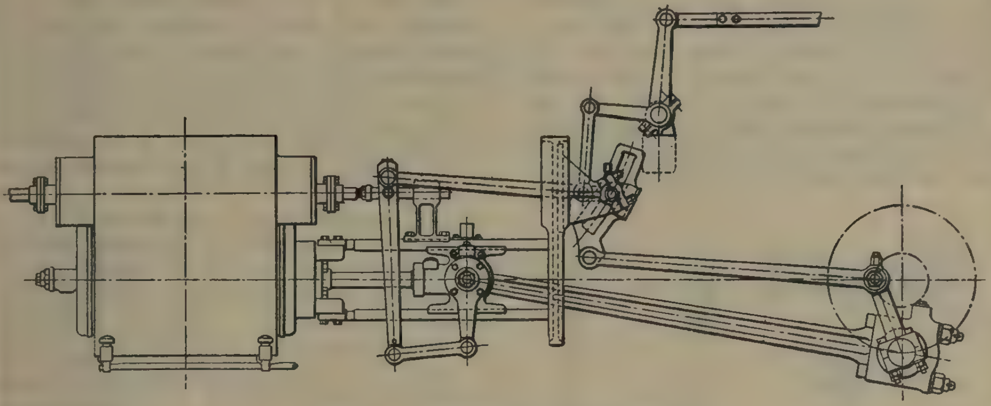
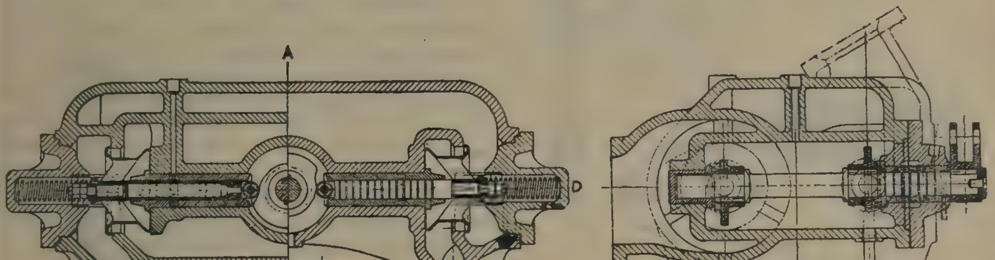


FIG. 26. WALSCHAERT'S VALVE GEAR



Section through Exhaust Port.

Section through Steam Port

Section on AB

Face of Flange

Section on CD

FIG. 27. LENTZ POPPET VALVES

In the Gooch motion, Fig. 23, the link is curved and stationary, whilst the connecting rod to the valve is moved up or down to receive the motion of the eccentrics. This motion was much used on the Continent of Europe, as it was claimed that a more constant and uniform distribution of steam was effected when the engine was running in different degrees of cut-off.

The Allan motion, Fig. 24, embodied a compromise between the Stephenson and Gooch arrangements; both the expansion link and valve spindle connecting rod were moved, one balancing the other; consequently, the link became a straight one. Although only used to a limited extent in this country, it became a great favourite in America, due to its ease of manipulation.

These so-called "link motions" have the disadvantage of slow movement for the valve during the whole of its path; manifestly, an ideal arrangement would be a quick opening of the port to steam, a period of little movement or rest whilst the steam is entering the cylinder, than a quick and effectual closing of the port, with a second interval of rest while steam completes its expansion in the cylinder. These desiderata were early recognized by designers of valve gears, and, to obtain the required movement, recourse has been made to different types of "radial" gear receiving motion from either reciprocating, or revolving, parts of the mechanism. The Walschaert is one of the earliest, dating as far back as 1844. Joy's preceded it, however, in extensive application in this country, hence its position in the list given.

In Joy's gear, Fig. 25, the movement for the valve is taken from the connecting rod, and no eccentrics are used. From the diagram given, it will be seen that the pin from which the valve spindle receives its motion moves in an elliptical path and will, consequently, perform the passage round the end arcs of the ellipse quicker than along the flatter, curved sides; hence, the speeding-up of the valve at the points of cut-off, release, etc. Reversing is effected by partially rotating the slipper block. An advantage of some importance is the absence of eccentrics and the consequent ability to straighten up the crank axle of an inside-cylindered engine, but there is the disadvantage of violent movement of the joints and moving parts, with accordingly much wear and tear.

The Walschaert gear must now be described, as it appears to be rapidly superseding nearly all other types of link, or radial, motion, chiefly on account of its simplicity, accessibility, and the very satisfactory distribution of steam which can be effected by it. The drawing, Fig. 26, shows an approved arrangement as installed in an outside-cylindered locomotive of recent construction. Only one eccentric, or crank, is used for each cylinder, and the travel of the valve is effected by a combination of movements of the cross-head and eccentric path.

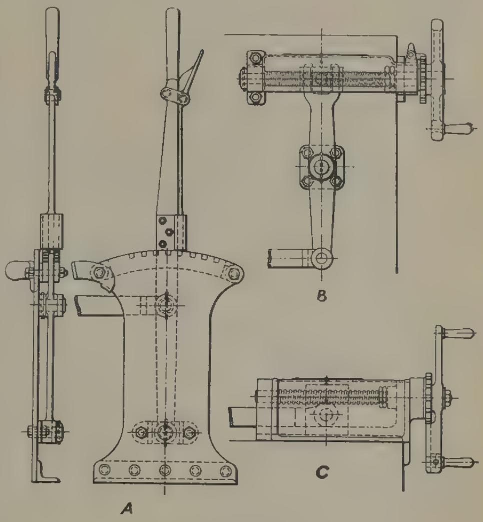


Fig. 28

In both Joy's and Walschaert's gear, the lead of the valve remains constant.

A still more recent improvement in locomotives is the introduction of "poppet" valves, actuated by reciprocating, or rotating, gear. thus securing the nearest result to that desired theoretically in the quick and definite opening and closing of the valves.

Fig. 27 shows one form of such a valve gear, which is being largely used on new locomotives, both here and abroad. It is known as the "Lentz," and another equally promising device is that of Caprotti. The valves in the former are actuated by cams operated by a cross-shaft in the valve chest, receiving motion from a modified radial valve gear operated by the driving members of the engine. In an arrangement of the second named, a horizontal revolving shaft, driven from the main axle, imparts the requisite motion to the cams used for operating the valves. It is claimed for this that the suppression of all links, reciprocating rods, etc., more than compensates for any complications due to the somewhat involved mechanism of the bevel wheel gearing, etc., employed.

For easy manipulation by the driver of the different valve gears, or motions, described, two distinct manual arrangements may be used, viz., levers, or screws; the latter is by far the more generally adopted on running locomotives, whilst the former has its advantages for shunting engines, where quick and constant reversal of the engines is of more importance than the ability to put and maintain the engine in varying degree of expansion. A representative application of each is shown in Fig. 28. *A* represents a lever arrangement, whilst *B* and *C* show screw installations.

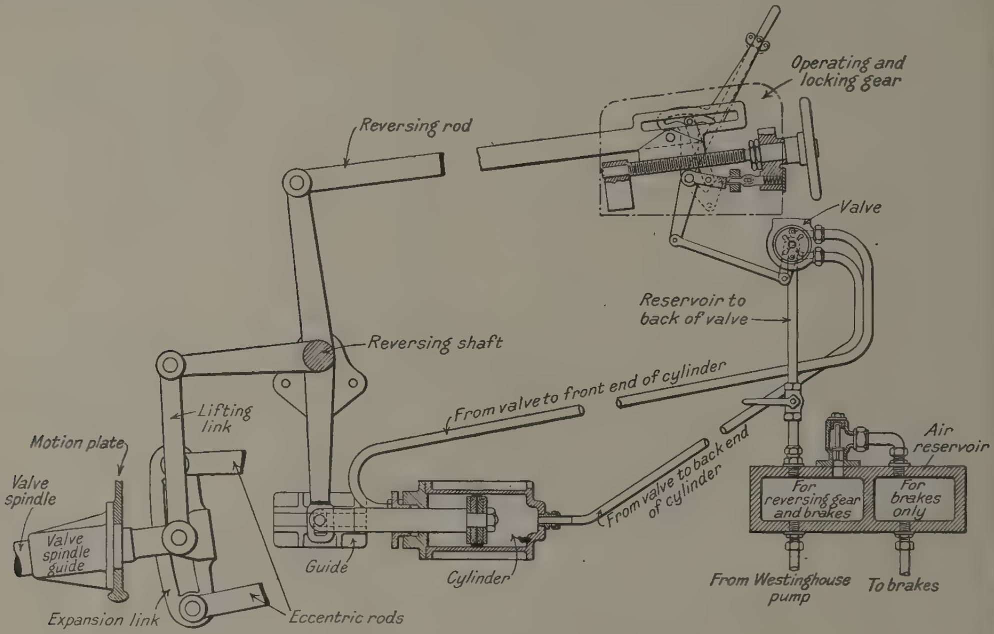


FIG. 29. AIR REVERSING GEAR

Power-operated reversing gears are, however, becoming general on engines of modern design, either steam or compressed air being used as the medium. Fig. 29 gives a very compact and satisfactory design of air-operated reversing gear, which can also be manipulated by hand when the engine is "dead," etc. In running, almost instantaneous reversal can be effected by power, whilst the finest adjustments for expansion can be produced by the hand-wheel.

STEAM BOILERS AND THEIR ACCESSORIES

By Percy E. Rycroft, M.B.E., M.I.E.E.

LESSON X

PULVERIZED COAL

The burning of coal in pulverized form has attracted a good deal of attention during recent years, and a number of plants have been installed for the purpose both in America and in this country.

There are now several types of such plant on the market, but these differ only in regard to detail, the main principle common to all being the introduction into a furnace of a continuous stream of finely powdered coal which, when ignited, maintains a gas-like flame.

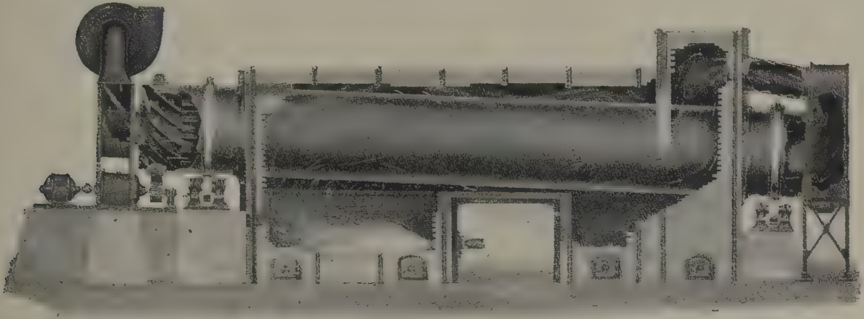


Fig. 60. Fuller Indirect-fired Rotary Drier—Pulverized Coal-fired

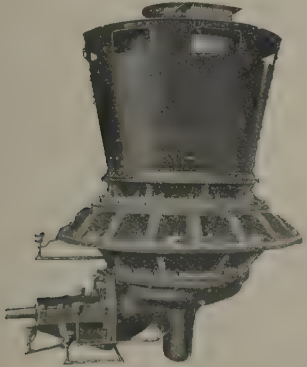


Fig. 61. Lehigh Pulverizer Mill, Air Separator Type

The chief advantages claimed for such an arrangement are—

1. The amount of excess air required for combustion can be reduced to a lower figure than is possible by other methods of coal firing, and a higher average efficiency can therefore be obtained.

2. The amount of heat delivered to a furnace can be so easily and quickly adjusted that a steam boiler fired by pulverized fuel has very great flexibility and a heavy overload capacity.

3. Fuels such as low volatile slacks and others containing a very high percentage of ash, which are difficult to burn satisfactorily by other means, can be readily consumed in a pulverized form.

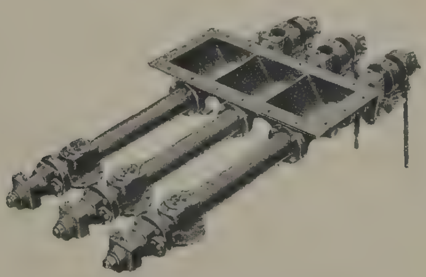


Fig. 62. Triplex Type Feeder (Fuller System)

There are two main systems of pulverized coal burning. In the " Central " system all the coal for a complete installation is dried and pulverized by a central plant and conveyed to the individual boilers through overhead bunkers, pipe lines, and feeders. In the " Unit " system each boiler is fitted with its own pulverizing mill and the fuel is delivered direct from this to the furnace.

At the moment, the " Central " system is most in favour and practically all the larger installations so far laid down are so arranged, although some success has been achieved by small " Unit " plants.

It is rarely possible to equip existing boiler plants for pulverized fuel firing without material modification to the furnace and, for the best results to be obtained, the steam-raising plant should be specially arranged for the purpose. The difference between the size of furnace required for a pulverized fuel fired plant and for a stoker fired plant of similar evaporative capacity is considerable, the former being at least three times as large as the latter.

The fact that higher temperatures can be developed in a pulverized fuel furnace would normally entail very heavy wear and tear on the brickwork, and to reduce the cost of maintenance, due to this cause, water cooled walls are now generally resorted to.

By this arrangement water tubes extend down both the side and rear walls and form an integral part of the boiler. A further difficulty which has to be contended with is the question of " slagging," i.e. the liability of the incombustible matter in the fuel to fuse into a solid mass under the high temperature and completely choke the ash pits.

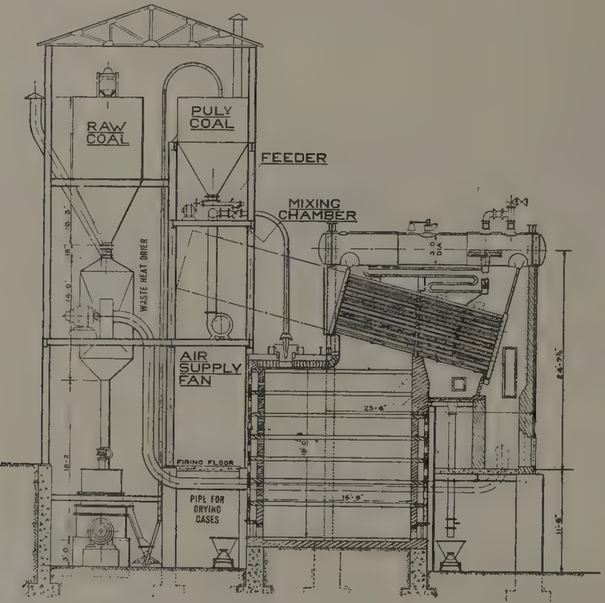


Fig. 63. Arrangement of Pulverized Coal-fired Boiler (Fuller System)

To overcome this trouble a horizontal screen of water tubes is often introduced a little above the bottom of the furnace, and by striking against this screen the white hot particles of incombustible matter are sufficiently cooled to prevent fusion. This water heating surface also forms an integral part of the steam-raising plant.

Another arrangement for achieving the same purpose allows for a stream of cold secondary air to meet the slag particles as they fall, and so reduce their temperature sufficiently to avoid fusing as they come to rest.

The two processes of primary importance in connection with pulverized coal burning are those of drying and pulverizing.

It will be obvious that any moisture in a powdered mass will cause it to bind and clog any mills, screens, pipes, elevators, bins, feeders, and other parts through which it has to pass on its way to the burners in the furnace, and this clogging is bound to be cumulative in its effects and may easily and speedily prevent an entire plant from operating.

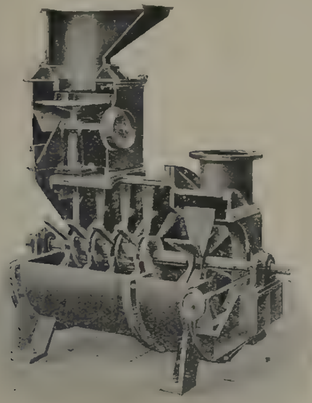


Fig. 64. Simon Carves Unit Pulverizer

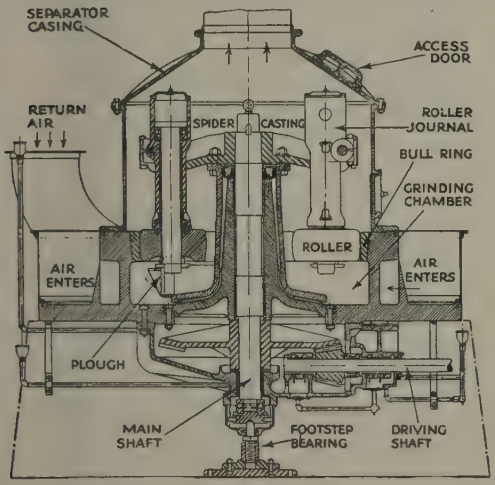


Fig. 65. Cross-section of Raymond "Super" Mill

For really efficient working, the moisture content of the fuel should, therefore, be reduced to not more than 1½ per cent, and with anything over 4 per cent difficulties are certain to be encountered.

Driers. Driers take two main forms, rotary and stationary. The former may consist of a steel cylinder, probably 50 ft. long and 6 ft. in diameter, carried in a cradle and slightly inclined to the horizontal.

It is slowly revolved by means of a pinion gear and the coal to be dried gravitates from the higher to the lower end, this motion being assisted by slats riveted to the interior which continually carry a certain amount of the coal upward until it falls back into the centre.

Either separately fired or waste furnace gases are passed both through and round the revolving drier, and by this means a high percentage of moisture is extracted from the fuel.

In the stationary or vertical type of drier the fuel enters at the top and slowly joggles through a series of baffle plates or louvres, while waste furnace gases pass through the fuel in the opposite direction of travel. Steam heated driers are also sometimes used.

Pulverizers. After drying, the fuel is passed into the pulverizing mills. These are of several types, but in all cases it is necessary to reduce the ground fuel to a high degree of fineness and, for really efficient working, it is considered that not less than 68–70 per cent of the bulk should pass through a 200 mesh sieve (i.e. 40,000 holes to the sq. in.), while the remainder should pass a 50 mesh.

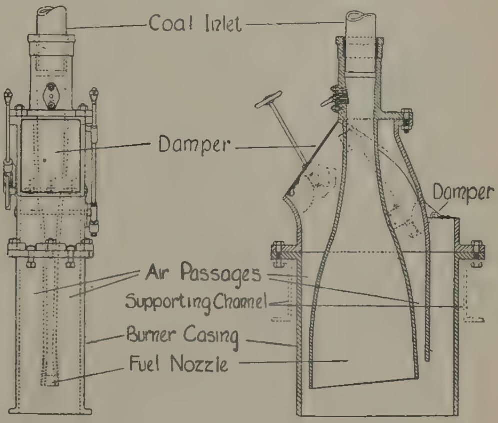


Fig. 66. Detail of "Lopulco" Burner

Of the mills, the ball mill as used in the Fuller system and the Raymond mill as adopted by the Lopulco system are typical. In the ball mill four heavy unattached iron balls are caused by rotating arms to travel round a wheel race to which the fuel is continually being fed.

As the coal is ground and crushed to a suitable fineness, it is carried by fan-induced air currents to a separating chamber and ultimately to the storage bins.

The Raymond mill is of the roller type consisting of six chilled iron rollers suspended on spindles and free to swing from a rotating spider driven by a vertical shaft. Surrounding the rollers is a pulverizing track or ring between which and the revolving rollers the coal is ground.

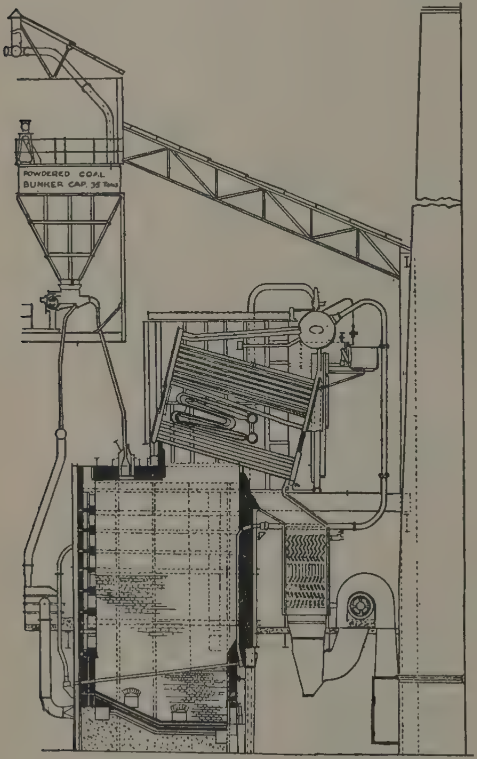


FIG. 67. CROSS-SECTION OF PULVERIZED COAL-FIRED BOILER PLANT (LOPULCO SYSTEM)

Given the dry fuel, of a suitable degree of fineness, the remaining operations for pulverized fuel burning are not difficult.

The fuel may be carried long distances—often many hundreds of yards—by means of a Fuller-Kinyon pump, in which a small amount of compressed air is allowed to mix with and agitate the powdered fuel so that it assumes a semi-fluid condition.

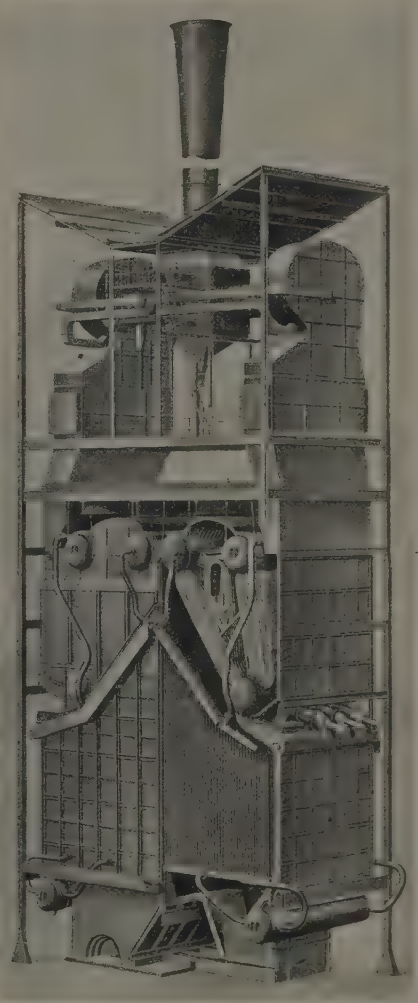


FIG. 68. SIMON CARVES BOILER ARRANGED FOR BURNING PULVERIZED COAL

Rated at 150,000 lb. per hour (180,000 lb. per hour on overload) 320 lb. pressure

It is stored in bins above the boilers and falls as required down enclosed chutes or pipes into mechanically operated feeders. These are usually of the screw feed type, and the fuel is carried forward until it meets a supply of primary air by the aid of which it is projected through the burners into the furnace.

The burners are arranged either horizontally or vertically, the former offering the advantage of reduced height of furnace, but the vertical burner is more frequently met with.

The burner flame is designed to present as much surface to the combustion air as possible and usually assumes a ribbon form, the combustion air being admitted through suitable regulators in the burner casings.

It will be evident that providing, as has already been indicated, the fuel is really dry and has been sufficiently finely pulverized, operation of the burners and feeders is a simple matter and capable of very ready adjustment. The amount of accessory plant required is high and, owing to the increased height of the combustion chambers required and other factors, the overall total cost of a steam plant equipped for pulverized coal burning is higher than for a stoker fired installation of similar capacity.

As the size of an installation increases this difference is less marked, and it is in the larger plants that pulverized coal firing shows to best advantage.

On the question of fuel a word of caution is advisable. It is true that there are numerous grades of coal which can more readily be consumed after pulverizing than in their raw form. This is not to say that by pulverization any coal can be made to burn equally well. Actually, coal containing less than 5 per cent of volatile matter is extremely difficult to burn and where, in addition to so low a volatile the ash content is high, continuous combustion becomes practically impossible, and a flame of very variable intensity is the result. Too much reliance should not, therefore, be placed on the suggestion that fuels, now a " drug on the market " and, consequently, to be obtained at a very low price, can be substituted for high grade fuel. This is at best a very transitory advantage because the amount of poor fuel available is a very small proportion of our total supplies, and even this speedily approximates in commercial value to the higher grades immediately a method of utilizing it is discovered.

PATTERN-MAKING AND FOUNDRY WORK

By J. McLachlan and C. A. Otto

LESSON XII

CORE-BOX CONSTRUCTION

The proper construction of core-boxes is as difficult as the construction of patterns. In some pattern-shops, when it is known that the core must be in two or three parts, only one core-box is made, and the core-maker is left to make his own divisions with the aid of paper or tin. This may be economical pattern-making, but it is costly core-making and seldom justifiable. Perhaps the only cores for which it is justifiable to make one core-box are some valve cores, because in these cases the valve seat forms a division, and two core-boxes are really constructed as one.

Much that has been said about pattern construction is equally applicable to core-box construction. In large boxes, the bottom and sides of which have to be made of several widths of timber battened together, joints should be open. There should be as few loose pieces as possible, and when some are necessary, except for " one-off " jobs, they should be dovetailed or dowelled in position in order that they may be accurately replaced. It is very important that, in rectangular cores, as much taper as possible should be allowed, and this applies also to ribs that are at right angles to the horizontal joint of the box.

Small Cylindrical Core-boxes. When a small cylindrical core-box is long, the grain will be as shown at Fig. 55, but when short it will be as Fig. 56. Assume that, instead of the core being of one diameter throughout its length, it is what is sometimes called a " bottle-necked " core. If the grain of the wood is almost perfectly straight there would be no great difficulty in cutting the shape; the best practice is to finish the small core-shape first; but if the grain is twisted, it is exceedingly difficult to get a satisfactory result. For this reason it is not uncommon, even with the long cores, to make the grain across the core shape. Sharper tools are required for cutting end grain, but a much better result is obtained and in less time.

For one-off jobs, core-frames are in many cases satisfactory, and, in some instances, when a number of cores are required, a core-frame or plate may be a more practical as well as a more economical way of obtaining a core than with a core-box. When determining the construction of core-boxes or frames, the pattern-maker must have an understanding of foundry practice. For instance, overhanging parts should be avoided in core-boxes whenever possible. It is frequently necessary, in order to prevent this, to print underneath a bracket or rib in a core-box and make a special core-box to form a core that is afterwards located in the main core.

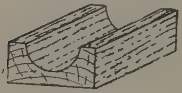


FIG. 55

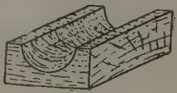


FIG. 56

FIG. 57

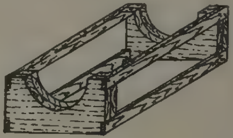


FIG. 58

In regard to cylindrical cores, Fig. 57 illustrates a satisfactory method of making a medium sized core-box when the shape is not more in length than, say, two-thirds of its diameter. If considerable taper could be allowed, it would be quite permissible to make a deeper box in this way. It is a usual type of box for a thin metal job, because any change that may take place will be uniform all round the diameter and will be small.

Fig. 58 shows the best method of constructing a box for a large cylindrical core, or even a medium sized cylindrical core which is long. Such a box should always be built, lagging being fitted to a framework consisting of grounds and distance stays. The grounds should be made of crossed thicknesses, if many cores are required, and not less than 2½ in. thick. The stays may be 2½ in. to 4½ in. wide, according to the size of the work, and 1¾ in. or 2 in. thick. The core-box should be built just slightly longer than the finished length of the core, in order that it can be cleaned up when the lagging has been fixed. The inside of the box has to be planed out finally to the core diameter. There is no danger of the framework becoming twisted if reasonable care is exercised when cutting the various checks. The grounds should be screwed together before being cut to shape, and the distance between the grounds should be 18 in. or less.

It is quite common practice, even with first-class work, when a full cylindrical core is wanted to make a half core-box from which half cores are made and afterwards pasted together, or, alternatively, to screw ends on the core-box, the top parts of which are semicircular and on which a strickle to form the top half of the core can ride.

Square and Rectangular Cores. In all pattern-shops, boxes for large and small rectangular cores are made, and, generally speaking, they do not present much difficulty. With very small cores it is best to make a box like Fig. 59, making a diagonal joint, because the core is more easily removed from such a box. It is the usual practice when many cores are required to make a multiple box, and sometimes metal boxes are provided in such cases.

Only the smallest boxes can be made in the way just described. Thin slab cores may be cut from a thickness of hardwood, but the great majority of square and rectangular cores are made in framed core-boxes. The thickness of timber for framed core-boxes should be from 1 to 1½ in., depending on the size of the box. Cheese-headed screws with washers are very suitable for framed core-boxes, which the core-maker has to take to pieces. A bottom may be an advantage or it may be a disadvantage. If the box has no bottom the core-maker can make the core direct on a plate, and all the woodwork can be removed before the core is transferred to an oven to be dried. Often, however, bosses and ribs have to be fastened to a bottom. With other than the smallest boxes, bottoms should not be screwed from the under-side, but by blocks about 1½ in. square and about 4 in. long—buttons they are sometimes called—which are secured to the bottom, the sides and ends of the box, as in Fig. 60. This obviates the trouble of removing bottom screws when stripping the core.

Very large and deep framed boxes are made somewhat differently from small boxes. It may be necessary to make up the sides of two or three different widths battened together. A plan of a suitable construction is shown in Fig. 61. It will be observed that the end-battens of the sides are screwed on the inside, and serve the same purpose as checks on smaller boxes.

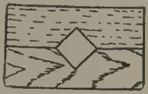


FIG. 59

Boxes for Irregular Shapes. It is sometimes very convenient when making boxes for irregular shaped cores to make a frame and fill in with shaped blocks. It is also done sometimes when making boxes for cylindrical cores about a foot diameter, with an end-fitting on a circular body. The construction of such a box is made apparent at Fig. 62. Sometimes instead of filling-in, the sides are made thicker and shaped.

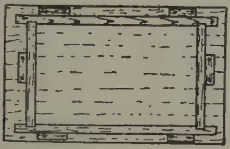


FIG. 60

Sectional Boxes. The cores for many circular castings are made in sections. Ribs may determine whether the sectional core-box shall be a sixth or an eighth, or the diameter may be the determining factor. Even when the centre boss is of fairly large diameter, it is always better to make the box to the centre as shown at Fig. 63, as it is the more easily altered, and bosses, ribs, etc., can be set more easily because the centre is on the box. The circular part in the case of large boxes should be built segmentally. It is frequently necessary to shape the top of a core like this, and it may be done by means of a strickle swinging from a centre dowel and resting on the circular part of the box, the sides being shaped by the strickle.

In regard to skeleton core-boxes, the core shape should always be built inside a framed box. For small work this outer box need not be more than 1 in. thick, but for large boxes it should be at least 1¼ in. thick, and in some cases even thicker. For small boxes the ends should be housed into the sides, but when the work is long and deep, say, 24 in. or more in length, it is quite satisfactory to dispense with the housing and use battens and check battens.

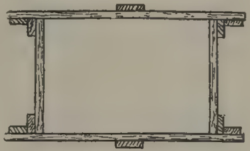


FIG. 61

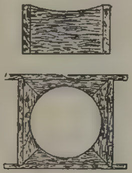


FIG. 62

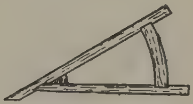


FIG. 63

The first thing to be done when making a skeleton core-box is to prepare a bottom, or two bottoms, if a full core-box is being made that necessitates two parts. It may save time ultimately to draw a very full view of the core-box joint on the bottom, because, if each part has to be fitted separately, it can be tested before being screwed or nailed into position. All skeleton core-boxes should have joint-plates, and these should fit inside the outer frame and should rest on the section pieces which define the core shape.

SOLDERING, BRAZING, AND WELDING

By E. Arthur Atkins, M.I.Mech.E.

LESSON III

TYPES OF JOINTS FOR ELECTRIC ARC WELDING

One of the most important things in connection with arc welding is to see that the work is properly prepared, to ensure so far as possible that the best kind of joint can be made.

The exercise of care and forethought in fixing up the work will always be well repaid in the higher quality of the result. Consideration should be given to the heating up of the metal, its welding, and subsequent cooling; and if the results of these are taken into account, it will go a long way towards producing a good job.

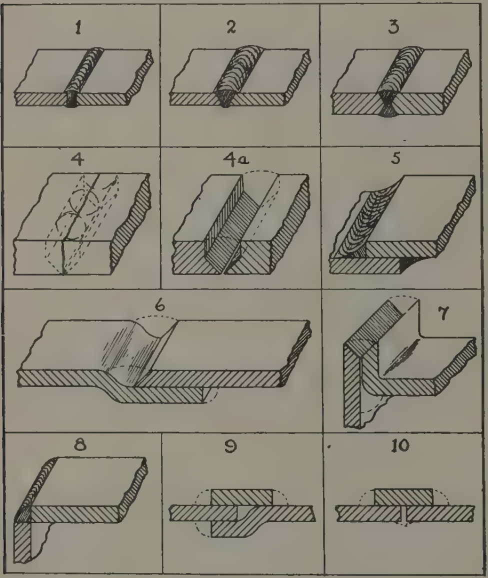


Fig. 9. Forms of Welded Joints for Plates

It should be remembered that the local heat of the electric arc is more intense than in other welding processes, and also that the heat diffusion is, as a rule, considerably less.

The speed at which the metal can be deposited is also more rapid than is the case with gas welding.

It is necessary that the above facts should be taken into consideration when arranging the work to be welded. It will, therefore, be seen that very much depends upon the intelligent foresight of the operator as to whether the resulting weld will be a success or a failure.

The important points to consider which go to the making of a good weld are as follows—

1. The operator should pass through a period of training in the making and testing of the various types of joints.
2. The welding machine should be of the best manufacture suitable to the work required to be done.
3. Joints should be so prepared that proper penetration can be obtained. Also the surface of the metal to be welded should be properly clean.
4. The electrodes chosen, whether bare, flux-coated, or lagged, should be suitable for the material to be welded.

The expansion and contraction of metals through increase or decrease of temperature gives a good deal of trouble in welding work, and the conditions set up in this way must be taken into account. No definite rules in this respect can be given, as each particular piece of work must receive careful consideration. The one important point to remember is that the metal of the work should not be left in a strained condition when cold.

Generally, plates less than $\frac{1}{8}$ in. thick need not be veed, but it is advisable in thin material to leave the joint slightly open so as to obtain good penetration.

For large work, the need for preparation is greater, and care should be taken to see that everything required is at hand, so that once welding is commenced there should be no stoppage on account of anything being required which might have been arranged beforehand.

Methods of Jointing

Plates may be connected together by electric arc welding in a variety of ways, and the operator by practice will soon learn which type of joint to apply to a particular class of work so as to obtain the best result.

Forms of joints are shown in Figs. 9, 10, and 11.

Joint for Thin Plates. No. 1 is suitable for $\frac{1}{8}$ in. or thinner plates. In this it will be seen the ends are left slightly open to allow for the deposited metal to be completely fused to the walls through the full thickness.

Joint for Medium-thickness Plates. No. 2 shows the single vee joint, which is suitable for plates from $\frac{1}{8}$ in. to $\frac{3}{8}$ in.

Joint for Thick Plates. No. 3 is the double vee joint for plates thicker than $\frac{3}{8}$ in. The edge of the plates may be prepared by either grinding or cutting away by other means. With a double vee it is, of course, necessary that the joints should be welded from both sides.

Welding a Cracked Plate. Nos. 4 and 4(*a*) show the preparation of a joint for welding a cracked plate. Where possible, the joint may be machined or drilled out as is most convenient, the point of the drill just penetrating the bottom of the crack, the division pieces in between being removed with a cross-cut chisel. This method of preparation is suitable for cast-iron work, especially if there is any danger of the crack extending.

Simple Lap Joint. No. 5 illustrates the form of the simple lap joint on which no preparation is needed, the end of one plate being welded on to the flat of the other. In connection with this joint, great care must be taken that the plate is not thinned by too much fusion at the edge of the filling, as this may result in pitting and cause considerable weakening of the plate.

Joggled Lap Joint. No. 6 is a joggled lap joint, which may be used when it is necessary to have one side of the joint perfectly flush with the plate surface. It will be noticed that it is necessary to chamfer the edge of one plate only.

Joints for Tank Work. No. 7 is a type of joint which may be used in fastening bottom plates for tank or drum work. This joint may be welded on the outer edges alone, which is generally sufficient, but if required and is get-at-able on the inner side, as indicated by the dotted curved line.

No. 8 shows the simple method of forming a right-angle vee corner joint without the plates being prepared. This makes a simple and quite good connection for tank construction or for boxes. In the making of this joint, it is necessary to have a knee-plate fixed or some other method for holding the plates in position.

No. 9 shows the construction of a joggled joint with cover strap. This method of jointing, which is rather unusual, gives no increased strength above that of No. 6; its chief advantage, however, is that it adds considerable stiffness to the plate and can be used in connection with welded ship construction.

No. 10 is an ordinary butt and strap joint, which can be still further stiffened by making a 60° vee where the plates butt together. This is, of course, a less costly joint to make than No. 9, but has not quite the same stiffness.

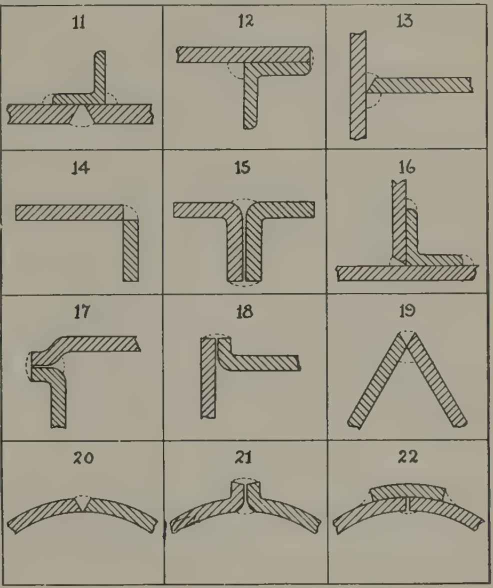


Fig. 10. Welded Joints

Extra Stiff Joints. Where considerable lateral stiffness is required, the method of construction shown by No. 11 can be adopted. Whilst it is necessary to weld the two plates together at the vee throughout the whole length the angle iron may be attached by tacking at intervals of about 4 in. to 6 in. This method of jointing simply takes the place of a riveted stiffener.

No. 12 shows a method of securing an angle bar to a plate to form a tee section, in case it is impossible to obtain the required size of tee iron to carry out the particular job.

No. 13 shows another method of forming a tee section.

No. 14 shows how two flat bars may be joined to form an angle.

No. 15 shows how two angle iron bars may be joined to form a tee section.

No. 16 illustrates how two plates may be jointed at right angles and reinforced with an angle bar.

Nos. 17 and 18 show methods that can be adopted respectively for welding the tops and bottoms of closed tanks.

No. 19 shows how plates can be welded together at any desired angle.

No. 20 is the 60° vee method which can be used for the side seams of a tank.

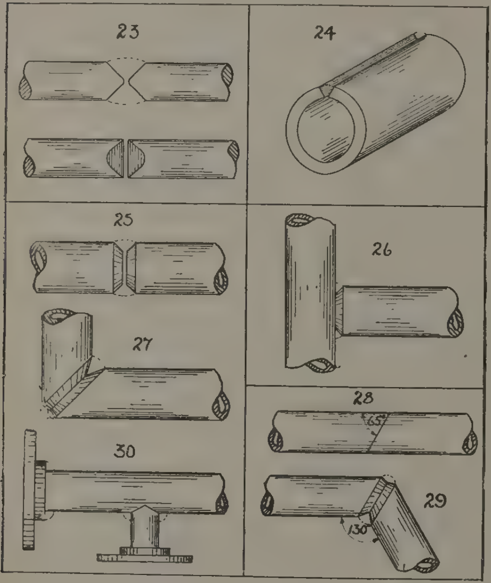


Fig. 11. Welded Joint for Bars and Pipes

No. 21 shows a flat side seam for a tank where extra stiffness is required.

No. 22 shows a butt seam for a cylindrical tank, together with curved cover plate.

No. 23 shows how the two ends of a round bar may be jointed. The ends of the bars should be ground to chisel edges and not with conical ends, as with the latter great difficulty will be experienced in depositing the metal, this running around the joint and simply sticking on instead of being welded.

Pipe Joints. No. 24 shows the kind of joint which can be used for pipe work. In this the vee can be made about 60°, but when the plate is more than $\frac{3}{4}$ in. thick the method shown in No. 4 joint can be used with advantage.

No. 25 is the method that is commonly used when welding together the ends of two pipes.

No. 26 illustrates how a branch pipe can be welded on to a main pipe to form a tee, the joint in this being reinforced according to the pressure the pipe is required to stand.

No. 27 shows how two pipes can be jointed together to form a right angle elbow.

No. 28 shows the method by which a pipe can be sawn so that the two parts can be joined together to form an elbow of any required angle. Thus, if a branch is required to be 130°, as shown in No. 29, it will be necessary to saw the pipe off at an angle of one-half the 130°, that is, 65°, as shown in No. 28. In joining the two arms are simply reversed, the two 65° angles together making up the bend of 130°.

No. 30 illustrates the way in which flanges of any thickness can be welded on to the ends of pipes.

Setting Up Work

One of the most difficult things to do in welding is to produce work that will be absolutely correct both in size, shape, and bevel; and to attain anything like a degree of accuracy, it is necessary to set the work up carefully and tack it every few inches before proceeding to carry out the main welding operation.

Where the longitudinal seam of a cylindrical article is to be welded, it is generally a good plan to drive a wedge into the joint at the farthest end and commence welding at the other end. This opening of the joint at one end will allow for contraction to take place without distortion, if the wedge be removed at the right time.

Another plan sometimes adopted in the case of heavy work is to have an assistant to keep the joint open with a bar as the welding proceeds along, thus putting the edge of the plate in tension, which will balance the strain set up by contraction.

Movement of the Electric Arc

There are several kinds of movements of the electrode which can be used in connection with welding, and a good deal of practice is required before the operator obtains the skill which enables him just to have that exact "touch" which is necessary to carry out all kinds of work in the best possible manner. The various movements of the arc which can be applied to different classes of work are illustrated in Fig. 12.

A. This shows the half-circular movement which can be used on thin sheet and also for filling holes in thin metal. The speed of travel will be roughly inversely proportional to the thickness of the metal.

B and C. These diagrams illustrate the circular movement. Great care must be taken to avoid the bottling up of slag, and in the case of heavily coated electrodes it is not advisable to adopt this movement on account of possible slag inclusions.

D. This, as will be seen from the Fig. 12, is a pendulum-like movement which, in practice, gives good results. This method can be adopted when heavily coated rods are used, as the slag can be kept separate from the molten metal.

E. This movement, which it will be seen follows the form of three sides of a square, is a good one for saving the edges of the parent metal from being unduly melted away. It is also a good movement for welding cast iron with special coated rods, as it gives the movement required to puddle the added metal, which is essential for good welding.

F. This, as will be seen from the sketch, is a half-circular movement with a greater forward travel than that shown by *D*.

No. 1 and No. 2 (Fig. 12) show end views of the wrong and right methods of depositing metal. No. 1 shows two runs with the wrong method, and it will be noticed there is a space between the two runs which is not properly filled. No. 2 shows the correct method to adopt. After the first run is deposited, the centre of the second run should be made to come on the edge of the first run, then the centre of the third run arranged to come on the edge of the second run, in this way the full width of the deposit being made up.

Perpendicular Welding

In vertical welding it is generally a good plan to commence with the bottom run first, then the second, and so on until the part required to be done has been filled. The direction of travel can be chosen from *G*, *H*, or *I* (Fig. 12). If only a small amount of deposited metal is required, the method of movement shown by diagram *G* should be adopted; but if a large amount is required, then either of the movements *H* or *I* will probably be found more suitable.

It is important not only to have the correct current, but also to let the flow of the deposited metal control the speed of travel. Regularity in movement is essential if the metal is to be deposited evenly. It is hardly necessary to add that a fair amount of practice is required for a welder to become expert in vertical work.

Overhead Welding

Overhead welding is, of course, much more difficult than either horizontal or perpendicular work, and requires a good deal of thought and consideration. As the metal on the end of the electrode becomes molten, it, of course, tends to fall away, and this makes the process much more difficult and longer. However, it is surprising how a good welder can overcome these difficulties and succeed in projecting the metal so as to obtain an overhead deposit.

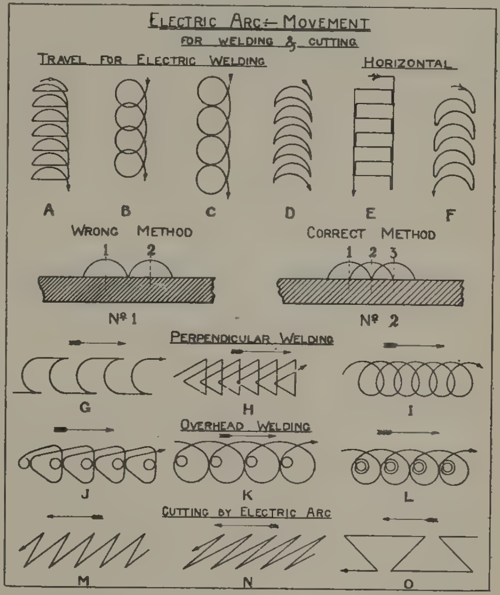


Fig. 12

The current required is usually a little higher than that ordinarily used, and special electrodes are sold which are particularly suitable for overhead work.

The face and hands must be well protected, and a covering used to safeguard the clothing when working in this awkward position.

As would be expected, it is a good plan to have the greatest heat at the centre of the vee, as capillary attraction will then have a better chance to operate and attract the metal to the surface to be welded.

Nearly every welder will have his own method of carrying out overhead work, and use that movement which he finds will give the best result; but as a little guidance, the three diagrams shown on Fig. 12 should be studied.

J. This shows the heat first in the centre, round one side, and back to the centre again before the movement is made to the opposite side of the vee.

K. Shows the heat in the centre, then a complete swing round.

L. Shows the movement which will give the greatest heat in the centre of the vee. The point of the electrode should be held almost at right angles to the work and, when the deposit has once started, adjustment made as required to suit the progress. It should always be kept in mind that the deposited metal will always tend to follow the liquid metal previously deposited. In other words, the molten metal will follow the arc.

It need hardly be said that a large amount of practice is essential if good overhead work is to be done, and experiments should be made on specimens of work that are of no particular value.

METAL CUTTING

Metal can be cut with a metallic arc, but the operation is quite different to that of the oxy-acetylene blow-pipe. In the case of the blow-pipe, the metal is oxidized away; whereas with the metallic arc, the only way by which cutting can take place is by actual melting. The use of a coated rod is preferable, because this increases the temperature of the arc. The travel is usually the opposite way to that for blow-pipe cutting.

Diagrams *M*, *N*, and *O* (Fig. 12) show three different motions of the electrode forcutting.

The zigzag motion should be applied as shown by diagrams, the method adopted being according to the size of cut and thickness of material. But, after a little practice, each operator will find out the movement which will give the best result for the particular job he is undertaking.

WORKS ORGANIZATION AND MANAGEMENT

By E. W. WORKMAN, B.Sc., A.M.I E.E., ETC.

LESSON XI

THE COST DEPARTMENT

General Organization. The place of the cost department in the scheme of organization varies greatly in different factories. In general, it is in charge of a " Works Accountant," who is responsible to the financial head of the business. Very often the cost department will include in its organization the time-keeping department, which is responsible for the calculation of wages, and the statistical department, responsible for the compilation of the controlling records. In an engineering factory where complicated piecework prices are paid for labour, the size of the department may be considerable—with corresponding possibilities for economy by means of good organization.

Objects of Cost Department. In a broadly-organized factory, the following may be said to be the main objects of a cost department—

1. To compile reliable records of what it has cost to make each article manufactured in order to provide a basis for fixing selling prices.

2. To examine past records critically in order to give the management information which will enable future policy to be decided in the most profitable way.

3. To provide statistical information which will cause waste of all descriptions to be reduced or eliminated.

4. To link up generally the work of the other

departments with the actual manufacturing, and act as a bureau of information.

Care must always be taken that the records kept in the cost department do not duplicate those kept in other departments, unless this is absolutely unavoidable.

What an Article Costs. The total cost of an article is made up of a number of factors, each of which requires very different treatment. These are as follows—

1. *The Material Cost.* This consists of the price paid for all the raw material in an article, together with the value of any parts bought outside complete. The proportion which the value of the material is to the whole cost naturally varies very much according to the nature of the product, but it may be said that it averages about 90 per cent in general machine work. In special cases, such as the moulding of castings, it will be very high, and in other cases where there is much complicated machine or fitting work it will be very much lower.

2. *The Labour Cost.* This consists of the wages paid to the employees for actually manufacturing the finished articles. It should be noted that the wages paid for making a part which has been manufactured with others in bulk really form part of the labour cost of the assembled article. In many cases the whole cost of manufacturing such a part (including the labour cost thereof) is included in the *material* cost, because of clerical difficulties in recording the labour incurred in making stock parts.

3. *The Cost of Overhead Charges.* The remainder of the total cost of an article consists of a proportion of all the varying general and particular expenses incurred in a factory and which cannot be conveniently charged direct to the article. Such expenses include rent, rates, insurance, power, light, heat, transport, depreciation, repairs, small tools, and the wages of indirect workers, such as firemen, inspectors, foremen, engineering staff, storekeepers, time clerks, and so on. In addition, there are many other charges, such as advertising, interest, and general selling expenses that have to be taken care of.

It will usually be found that the best organization of the cost department provides different sections to deal with each of these divisions of total cost, together with another section to co-ordinate the work of the other sections and compile the cost of the individual articles. In addition, there may be a separate section to deal with the many varieties of office machinery that it will probably be found advisable to install.

Sources of Information. A definite stream of information flows into the cost department to form the basis of the work which has to be done. In the case of material, a copy of all requisitions for material issued from stores should be sent to the cost department together with credit notes for any material returned to stores as not required. If not already priced by the stores department, these requisitions will require valuation according to the purchase price paid for the material. To ensure that the material cost is complete, it will be necessary to compare these requisitions with a detailed specification of the apparatus, a copy of which should be sent to the department when work is first commenced. This check is also necessary to exclude the possibility of duplicate parts being charged to the cost, due to scrapping of defective work. The cost of such defective parts should be charged to a separate account, so that the cost of different articles may be strictly comparable irrespective of special faulty conditions. By charging to separate accounts all defective work and material, these sources of loss are far more under control and not so likely to be lost sight of.

In the case of labour, the source of information is the job tickets described in the next lesson.

The total amounts of the overhead charges are obtained from the analysis of the actual amounts paid out, together with an estimate of payments outstanding. The proportioning of these overhead expenses to the different articles manufactured is a difficult and important matter, which has to be done with the greatest of care by one of several available methods.

Office Machinery. Probably in no other description of work can office machinery be put to such good use as in the cost department. Without the efficient use of office machinery it will often be impossible to obtain accurate costs without having an excessively large staff—with a correspondingly excessive overhead expense. At the same time, the use of office machinery in the cost department calls for the display of considerable organizing ability on the part of those responsible for its efficient use. The principal types of office machines and appliances which are commonly used in connection with costing work are as follows—

1. Adding and calculating machines.
2. Duplicating machines.

3. Addressing machines.
4. Numbering and dating machines.
5. Sorting and tabulating machines.
6. Time recorders.

The use of some of these machines applies more to the wages section than to the rest of the cost department, as for example, the time recorders and the addressing machines (used for the preparation of pay-rolls, pay packets, insurance cards, etc.).

In a large factory where a system of automatic sorting and tabulating is in use, the apparatus used is intricate, and the influence of the system is much more apparent in the general organization of the works. To get the greatest efficiency out of such an automatic system, it is important that the whole of the records and forms which affect the cost department in any way should be designed together with the special limitations of the machinery used kept ever in view. The detailed arrangement of the original documents—which in such a system have to be semi-automatically copied on to standard cards—will make or mar the success of the system. The other necessity is that of obtaining correct information on the original documents emanating from the shops.

Advantages of Automatic Sorting and Tabulating. In thinking of the clerical organization of the cost department in a large factory it must be remembered that, literally, millions of different documents may have to pass through the department in the course of a year. In such cases the advantages of an automatic system of sorting and tabulating are claimed to be—

1. The greatly increased speed with which returns and statistics can be compiled.
2. The increased amount of statistical information which is readily obtainable by rapidly re-sorting the card records into different classifications.
3. The elimination of hand-sorting and tabulation from the work of the clerical staff, with consequent increased attention to critical and constructive work.
4. A considerable saving of overhead charges as regards the item of office expenses for an equivalent amount of statistical information produced.

Information Given Out. It is the duty of the statistical section of the cost department to collect and issue to those concerned tabulated statements of all the activities of the factory. These statements should be prepared with a view to assisting the management to—

1. Find out what processes of manufacture are costly or inefficient.
2. Appreciate in what ways overhead expenses may be reduced.
3. Discover new and better methods of manufacture.
4. Know where time is being lost and material wasted, and thus find out how production may be speeded up.
5. Keep the factory full of work by selling at intelligent prices with a known margin of profit.

Returns Made. With the above objects in view, some of the returns which may be issued by the cost department are as follows—

1. Labour statistics, giving a comparative departmental analysis of the different pay-rolls.
2. Analysis of time spent in work by different classes of machinery, thus showing the distribution of the load.
3. Departmental analysis of the stores issued.
4. Analysis of the production of each department.
5. Analysis of the expenses incurred, due to defective design, defective materials purchased, or bad workmanship.
6. Comparative statements of departmental expenses.
7. Summary of the actual costs of articles manufactured, compared with the estimates prepared before manufacture.
8. Detailed analyses of the cost of selected articles.

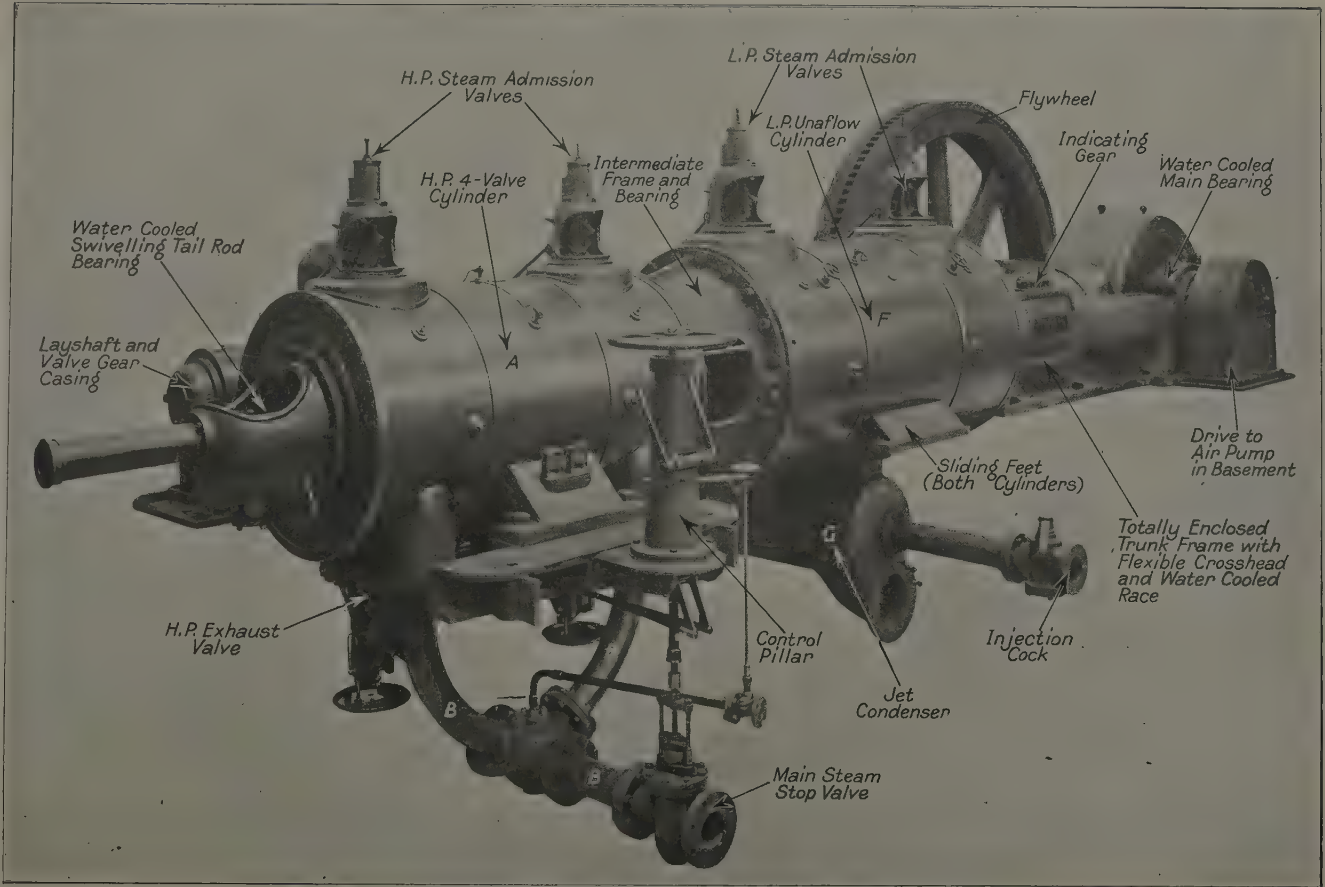

H.P. Steam Admission Valves
L.P. Steam Admission Valves
Flywheel
L.P. Unaflow Cylinder
Indicating Gear
Water Cooled Main Bearing
Intermediate Frame and Bearing
H.P. 4-Valve Cylinder
Water Cooled Swivelling Tail Rod Bearing
Layshaft and Valve Gear Casing
A
F
Drive to Air Pump in Basement
Sliding Feet (Both Cylinders)
Totally Enclosed Trunk Frame with Flexible Crosshead and Water Cooled Race
Injection Cock
H.P. Exhaust Valve
Control Pillar
Jet Condenser
B
Main Steam Stop Valve

FIG. 34. ARRANGEMENT OF HORIZONTAL STEAM EXTRACTION ENGINE
(See Article on "Reciprocating Steam-engines," page 1318)

MATHEMATICS FOR ENGINEERS

By W. G. Bickley, M.Sc.
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LESSON XXVII

SINES AND COSINES

In the lessons on trigonometry, we worked examples upon the displacement of a piston at various points of the stroke, in terms of the angle turned through by the crank, and of necessity employed sines and cosines of this angle. In practice, we are quite as much, if not more, interested in the piston speed and acceleration, and to find this we shall need to be able to differentiate the formula for the displacement, containing sines and cosines. Also, *any* case of oscillatory motion, and the study of alternating electric currents, will require the same mathematics, i.e. the differentiation of sines and cosines.

The fundamental formulae can be obtained either geometrically or analytically, and we shall give both methods. First, in dealing with sines and cosines, it is convenient to use a circle of *unit* radius, for if P is any point on such a circle (Fig. 110), and PM is perpendicular to OA, then

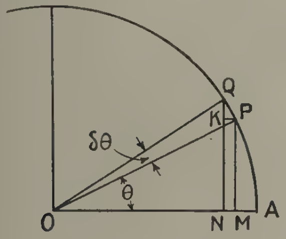


FIG. 110

$$PM = \sin\theta \text{ and } OM = \cos\theta,$$

θ being the angle POA. Also, if Q is another point on the circle near P, such that the angle POQ is $\delta\theta$ and QN is perpendicular to OA, we shall have

$$QN = \sin(\theta + \delta\theta) \text{ and } ON = \cos(\theta + \delta\theta)$$

Thus the increase in the sine, or $\delta(\sin\theta)$, is $QN - PM$, or drawing PK perpendicular to QN,

$$\delta(\sin\theta) = QK.$$

Again, since arc = radius × angle (in *radians*), and our radius is unit,

$$\text{arc } PQ = 1 \times \delta\theta = \delta\theta.$$

Thus,

$$\frac{\delta(\sin\theta)}{\delta\theta} = \frac{QK}{\text{arc } PQ}$$

But when $\delta\theta$ is a small angle (and we are going to make it tend to zero) the arc PQ and the chord PQ will be very nearly equal, and the smaller the angle, the nearer to 1 will their ratio be. Thus

$$\frac{\delta(\sin\theta)}{\delta\theta} = \frac{QK}{\text{chord } PQ} = \cos PQK$$

From the figure, it can be seen that the angle PQK is very nearly equal to θ, and that it tends to become more nearly equal to θ as $\delta\theta$ becomes smaller. (Actually, the angle PQK is $\theta + \frac{1}{2}\delta\theta$.) Using this, and proceeding to the limit as $\delta\theta \to 0$, we have, finally, the result sought,

$$\frac{d(\sin\theta)}{d\theta} = \cos\theta,$$

or, **the differential coefficient of a sine is a cosine** (the angle being in radians).

Dealing with the cosine in the same way, we have the increase of the cosine equal to $ON - OM = -NM = -PK$, so that

$$\frac{\delta(\cos\theta)}{\delta\theta} = -\frac{PK}{\text{arc } PQ} = -\frac{PK}{\text{chord } PQ}$$
$$= -\sin PQK,$$

and, proceeding to the limit as $\delta\theta \to 0$,

$$\frac{d(\cos\theta)}{d\theta} = -\sin\theta$$

or, **the differential coefficient of a cosine is** ***minus*** **a sine.**

Analytically,

$$\delta(\sin\theta) = \sin(\theta + \delta\theta) - \sin\theta$$
$$= 2\cos(\theta + \tfrac{1}{2}\delta\theta)\sin\tfrac{1}{2}\delta\theta,$$

so

$$\frac{\delta(\sin)\,\theta)}{\delta\theta} = \frac{2\cos(\theta + \frac{1}{2}\delta\theta)\sin\frac{1}{2}\delta\theta}{\delta.\theta}$$
$$= \cos(\theta + \tfrac{1}{2}\delta\theta)\,\frac{\sin\frac{1}{2}\delta\theta}{\frac{1}{2}\delta\theta}$$

Now as $\delta\theta$ tends to zero, the ratio $\dfrac{\sin\frac{1}{2}\delta\theta}{\frac{1}{2}\delta\theta}$ tends to 1, and $\cos(\theta + \frac{1}{2}\delta\theta)$ tends to $\cos\theta$, so we again deduce

$$\frac{d(\sin\theta)}{d\theta} = \cos\theta$$

The analytical proof of the formula for the differential coefficient of the cosine is left as an exercise.

Applications. 1. Knowing the sine and cosine of 45° (each is $\frac{1}{2}\sqrt{2}$ or 0·7071), find approximately the values of *sin* 46° and *cos* 46°.

Taking the sine first, if $y = \sin\theta$, we know that $y = 0{\cdot}7071$ when $\theta = 45^\circ$. Also, we have $\delta y = \dfrac{dy}{d\theta}\,\delta\theta = \cos\theta\,.\,\delta\theta$ (by the differentiation formula just proved). Now $\delta\theta$ is 1°, but this must be in *radians* before it is used in the formula, and $1^\circ = \pi/180$ radians. Therefore,

$$\delta y = \cos\theta\,.\,\delta\theta = 0{\cdot}7071 \times \frac{\pi}{180} = 0{\cdot}0123.$$

Consequently,

$$\sin 46^\circ = 0{\cdot}7071 + 0{\cdot}0123 = 0{\cdot}7194.$$

(Tables give 0·7193; had we used the *average* value of $\dfrac{dy}{d\theta}$ or $\cos\theta$, i.e. $\cos 45^\circ\,30'$, our result would have been better.)

Similarly, if $z = \cos\theta$, $\delta z = \dfrac{dz}{d\theta}\,.\,\delta\theta = -\sin\theta\,.\,\delta\theta = -0{\cdot}7071 \times \dfrac{\pi}{180} = -0{\cdot}0123$, and so

$$\cos 46^\circ = 0{\cdot}7071 - 0{\cdot}0123 = 0{\cdot}6948 \text{ (approx.)}.$$

Connected Rates of Change. The simplest (and fundamental) case of oscillatory motion is that termed Simple Harmonic Motion (S.H.M., as it is often abbreviated). It is usually defined as the projection of a uniform circular motion, i.e. if P in Fig. 111 describes the circle with uniform speed, and Q is the projection of P on the diameter, then the motion of Q is simple harmonic. If s denotes the displacement of Q from the centre O, and θ is the angle POA, we have

$$s = r\cos\theta,$$

r being the radius. Now, if ω is the angular velocity of P, we must have, with t representing time, $\theta = \omega t$, and so

$$s = r\cos\omega t.$$

To find the velocity and acceleration of Q we have to differentiate this with respect to t. We have here *three* quantities to consider, s, θ, and t, and require first the velocity $\dfrac{ds}{dt}$. We also know the angular velocity, which is represented by $\dfrac{d\theta}{dt}$, and is equal to ω. We can also find $\dfrac{ds}{d\theta}$ by our rule for differentiating a cosine—it is $-r\sin\theta$. We require a relation between these three differential coefficients—or rates of change—to enable us to find $\dfrac{ds}{dt}$ from the known values of $\dfrac{ds}{d\theta}$ and $\dfrac{d\theta}{dt}$; and this connection between three rates of change has other uses.

Follow the motion for a small interval of time δt. During this interval, the angle will increase by $\delta\theta$, and Q will move a distance δs. Now by the usual formula for small increments, we have

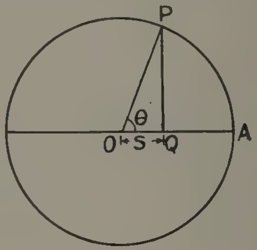


FIG. 111

$$\delta s \doteq \frac{ds}{d\theta}\,.\,\delta\theta$$

Now divide through by δt (to find the value of $\dfrac{ds}{dt}$, of course) and we have

$$\frac{\delta s}{\delta t} \doteq \frac{ds}{d\theta}\,\frac{\delta\theta}{\delta t}$$

As $\delta t \to 0$, this becomes more and more exact, and finally, in the limit,

$$\frac{ds}{dt} = \frac{ds}{d\theta}\,\frac{d\theta}{dt}$$

We can now use the values of $\dfrac{ds}{d\theta}$ and $\dfrac{d\theta}{dt}$, and find

$$v = \frac{ds}{dt} = -r\sin\theta \times \omega = -\omega r\sin\omega t.$$

It will be noticed that we have in this the result that

$$\frac{d}{dt}(\cos \omega t) = -\omega \sin \omega t$$

By the same method, if $y = \sin \omega t$, we shall have, putting $\omega t = \theta$, and consequently $y = \sin \theta$,

$$\frac{dy}{dt} = \frac{dy}{d\theta}\frac{d\theta}{dt} = \cos \theta \times \omega = \omega \cos \omega t$$

or

$$\frac{d}{dt}(\sin \omega t) = \omega \cos \omega t$$

Consequently, if a denote the acceleration in the simple harmonic motion,

$$a = \frac{dv}{dt} = \frac{d}{dt}(-\omega r \sin w t) = -\omega^2 r \cos \omega t$$

Noticing that $r \cos \omega t = s$, we have the most important feature of simple harmonic motion, namely that

$$a = -\omega^2 s,$$

or, in words, the acceleration is proportional to the displacement. For further treatment of simple harmonic motion, see the articles on *Applied Mechanics* and *Balancing of Engines.*

Applications (*contd.*). Another instance where we need to be able to differentiate sines and cosines is to follow the speed and acceleration in the delivery pipe of a reciprocating pump.

A pump has a cylinder diameter of 9 in., a stroke of 18 in., and it runs at 30 r.p.m. Find the speed and acceleration in the delivery pipe, if this is 3 in. in diameter and 40 ft. long, and also the piston pressure necessary to accelerate the water in the pipe.

Here, taking the motion of the piston as simple harmonic, the displacement s from the mid-point of the stroke is $9 \cos \pi t$ in. (The amplitude is half the stroke, or 9 in., and the angular velocity is π radians/sec.) If we work in feet, we shall have

$$s = \tfrac{3}{4} \cos \pi t \text{ ft.}$$

The piston velocity is here $-\frac{ds}{dt}$ (minus, because it is, at the commencement of the stroke, in the opposite direction to s). Thus,

$$v = +\tfrac{3}{4}\pi \sin \pi t \text{ ft./sec.}$$

If V is the corresponding velocity of the water in the delivery pipe, the velocities will be inversely proportional to the areas of pipe and cylinder cross-sections, i.e.

$$\frac{V}{v} = \frac{9^2}{3^2} = 9,$$

so that

$$V = 9 \times \tfrac{3}{4}\pi \sin \pi t$$
$$= 21{\cdot}3 \sin \pi t \text{ ft./sec.}$$

The acceleration of the water in the pipe is $\frac{dV}{dt}$, and so

$$a = \frac{dV}{dt} = 21{\cdot}3\,\pi \cos \pi t$$
$$= 67{\cdot}9 \cos \pi t \text{ ft./sec.}^2.$$

The pressure to accelerate a column of water l ft. long is lwa/g, w being the density (62·4 lb./cub. ft.), so that here

$$p = \frac{40 \times 62{\cdot}4 \times 67{\cdot}9}{32{\cdot}2} \cos \pi t$$

$$p_{max} = 5260 \text{ lb./sq. ft.} = 36{\cdot}6 \text{ lb./sq. in.}$$

(We have taken rather a small pipe diameter, and so get a high pressure; the pressure becomes negative at the end of the stroke, and cavitation would be likely to occur.) There is also, in practice, friction loss to be taken into account, depending upon the square of the speed. This varies during the stroke, and to find its total amount we must integrate; for this see the next lesson.

Crank-slider Mechanism. In dealing with reciprocating engines, the motion of the piston is of great importance. In Fig. 112, AB is the crank, BC the connecting rod, C the crosshead. The motion of the piston is the same as that of the crosshead, and if DE is the stroke of the crosshead, O being its mid-point, we have $AO = l$, l being the length of the connecting rod. Measuring the displacement of C from the mid-point of its stroke, and denoting it by s, we have

$$s = AC - AO = AC - l.$$

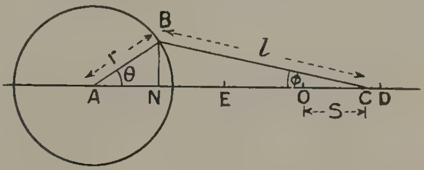


FIG. 112

Calling the length of the crank r, $\angle BAC\ \theta$, and $\angle BCA\ \phi$,

$$AC = AN + NC = r\cos\theta + l\cos\phi$$

But $\quad BN = r\sin\theta = l\sin\phi$

or $\quad \sin\phi = \frac{r}{l}\sin\theta$

Consequently $\quad \cos\phi = \sqrt{1 - \frac{r^2}{l^2}\sin^2\theta}$

$$= 1 - \tfrac{1}{2}\frac{r^2}{l^2}\sin^2\theta - \dots$$

by the binomial theorem, the other terms being small when $\frac{r}{l}$ is—if r is $\frac{1}{3}l$, and it is seldom so great, the error is 1 in 650 approx., at the worst point, when $\sin\theta = 1$; the approximation is quite good enough for any practical case. Remembering that $\sin^2\theta = \frac{1}{2}(1 - \cos 2\theta)$, we have

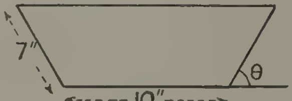


FIG. 113

$$\cos\phi = 1 - \tfrac{1}{4}\frac{r^2}{l^2} + \tfrac{1}{4}\frac{r^2}{l^2}\cos 2\theta \dots$$

Using this value of $\cos\phi$, we find

$$s = r\cos\theta + l\cos\phi - l$$

$$= r\cos\theta + \tfrac{1}{4}\frac{r^2}{l}\cos 2\theta - \tfrac{1}{4}\frac{r^2}{l}$$

The speed is $\frac{ds}{dt}$, and, since $\frac{d\theta}{dt} = \omega$, the angular velocity of the crank,

$$\frac{ds}{dt} = \frac{ds}{dt}\frac{d\theta}{dt} = \omega\frac{ds}{d\theta}$$

$$= -\omega r\sin\theta - 2\omega \times \tfrac{1}{4}\frac{r^2}{l}\sin 2\theta$$

i.e. $\quad v = -\omega r\left(\sin\theta + \tfrac{1}{2}\frac{r}{l}\sin 2\theta\right)$

(Notice that $\frac{d}{d\theta}(\cos 2\theta) = -2\sin 2\theta$, by the same rule.) The second term in brackets gives the effect of the obliquity of the connecting rod, which makes the actual motion deviate from simple harmonic. To find the acceleration, we have to differentiate again. In the same way,

$$a = \frac{dv}{dt} = \omega\frac{dv}{d\theta} \text{ and so}$$

$$a = -\omega^2 r\left(\cos\theta + \frac{r}{l}\cos 2\theta\right).$$

In this we see that the "obliquity" term is twice as important in the acceleration formula as it is in the velocity formula. The student should plot graphs showing s, v, and a, for various values of the ratio r/l (say, $\frac{1}{3}$, $\frac{1}{4}$, $\frac{1}{5}$, . . .) and see the effect of the connecting rod.

Maxima and Minima. A strip of metal 24 in. wide is to be bent into a trapezoidal channel section, the base being 10 in. wide and each side 7 in. Find the angle at which the sides should slope for maximum sectional area.

Calling the angle θ (Fig. 113), it will be found that the area is $A = 7\sin\theta\,(10 + 7\cos\theta) = 70\sin\theta + 24{\cdot}5\sin 2\theta$. ($\sin\theta\cos\theta = \frac{1}{2}\sin 2\theta$). For a maximum, we must have $\frac{dA}{d\theta} = 0$, so, $\frac{dA}{d\theta} = 70\cos\theta + 49\cos 2\theta = 0$.

Putting $\cos 2\theta = 2\cos^2\theta - 1$, and re-arranging,

$$98\cos^2\theta + 70\cos\theta - 49 = 0,$$

a quadratic equation for $\cos\theta$, giving, when solved,

$$\cos\theta = 0{\cdot}4350 \text{ or } -1{\cdot}1493.$$

The first gives $\theta = 64^\circ\ 13'$, and the second is impossible.

Solution of Equations. If we wish to divide a circle into three portions of equal area by two parallel chords (Fig. 114), we shall find that, calling the angle $AOC\ \theta$,

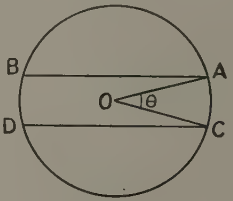


FIG. 114

$$\theta + \sin\theta = \tfrac{1}{3}\pi.$$

An approximation to this is $\theta = \pi/6$, and to get a better, put $y = \theta + \sin\theta - \frac{1}{3}\pi$. When $\theta = \pi/6$ ($= 30^\circ$), we have

$$y = 0{\cdot}5236 + 0{\cdot}5000 - 1{\cdot}0472 = -0{\cdot}0236,$$

therefore $\delta y = +0{\cdot}0236$.

(*Continued on page* 1312)

APPLIED MECHANICS

By Geo. W. Bird, Wh. Ex., B.Sc., A.M.I.Mech.E., A.M.I.E.E.

LESSON XXIII

Mechanics of Fluids

Fluid. A substance which offers very little resistance to forces tending to change its shape, and which takes the shape of the body with which it is in contact is termed a *Fluid*. All the fluids with which the engineer is called upon to deal—water, oil, air, and other gases—offer some little resistance to deformation, and this deformation, when it consists of a sliding of one part of the fluid over another (just as we imagine shearing to take place in metals), is termed *Viscosity*.

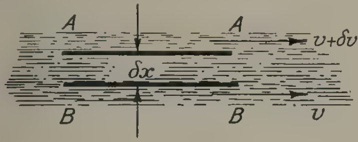


Fig. 144

Let us suppose that we have a quantity of fluid in motion, and let us imagine two planes AA and BB which are parallel to the direction of flow and at a distance apart δx, Fig. 144. The *law of viscous force* can be stated as

$$\text{Viscous force} \propto \frac{\delta v}{\delta x};$$

that is, the tangential force on any plane is proportional to the change of velocity and inversely proportional to the distance between the planes considered;

or,
$$\text{Viscous force} = \mu \cdot \frac{\delta v}{\delta x};$$

μ is termed the coefficient of viscosity.

Density. The density of a body is stated as the mass per unit volume, and for water at 4° C. (the temperature of maximum density) is about $62\frac{1}{2}$ lb. per cub. ft.

Pressure at Any Depth

Consider any area a whose centroid is at a depth h below the free surface of the fluid, Fig. 145. We may consider that this area a is supporting the column of water immediately above it, and hence is subjected to a load $= a \times h \times w$, where $w =$ weight per unit volume of the fluid; the pressure per unit area is, therefore, wh.

Total Pressure on an Immersed Surface

Let us consider any area ABC which may have any inclination as shown in Fig. 146. On any elementary area δa, situated at a vertical depth x below the free surface of the liquid, the intensity of pressure $= w \times x$, and the total thrust on this area $= w \times x \times \delta a$. The sum of all such elementary thrusts will equal the total thrust or pressure on the whole area, or, total pressure on whole area $= \Sigma\delta a \times x \times w$.

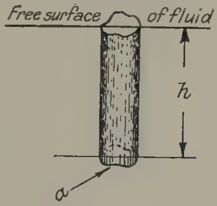


Fig. 145

But, $\Sigma\delta a \times x$ is equal to the whole area $(A) \times G$, the distance of the centroid of the area below the free surface, hence the total pressure on an immersed area

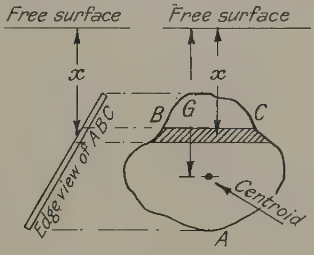


Fig. 146

$$= A \times G \times w.$$

Relation Between Head of Water and Pressure

Let us now consider the fluid water and obtain a relation between the head at any depth and the corresponding pressure. Take a box of the form and dimensions shown in Fig. 147 and fill it with water. The total load on the base

$$ABCD = 1 \text{ ft.} \times 1 \text{ ft.} \times 2{\cdot}3 \text{ ft.} \times 62{\cdot}5 \text{ lb. per cub. ft.}$$

$$ABCD = 144 \text{ lb. (approx.)}$$

and, as the area of the base $ABCD$ which supports this load is 1 sq. ft. = 144 sq. in., we

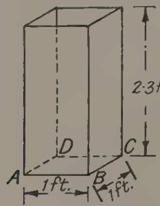


FIG. 147

see that every 2·3 *ft. head of water is. approximately, equivalent to a pressure difference of* 1 *lb. per sq. in.*

If the fluid is other than water, then the head required to produce unit (1 lb. per sq. in.) pressure difference $= \frac{2{\cdot}3}{\rho}$ ft., where ρ is the specific gravity of the fluid.

EXAMPLE. A hollow cone, diameter of base 10 in., height 12 in., is filled with water. Determine the total thrust on the base when the axis of the cone is (1) horizontal, (2) vertical, the cone resting on its base.

Case 1. Total thrust = area of base × depth of centroid of base below the free surface of the water × weight of unit volume of water,

$$\text{or, total thrust} = \frac{d^2\pi}{4} \times \frac{d}{2} \times \frac{62{\cdot}5}{1728}$$

$$\text{total thrust} = 14{\cdot}2 \text{ lb.}$$

Case 2.

$$\text{Total thrust} = \frac{d^2\pi}{4} \times \text{height of cone} \times \frac{62{\cdot}5}{144}$$

$$\text{or, total thrust} = 34{\cdot}08 \text{ lb.}$$

CENTRE OF PRESSURE

Since the intensity of fluid pressure varies with the depth below the free surface of the fluid, when we consider an immersed area which may be inclined at any angle θ to the vertical, we find that the intensity of pressure is not uniform, and we now require to find that single point at which we may consider the whole fluid thrust to act; this point is termed the *centre of pressure.*

Let E, Fig. 148, be the centroid of any plane surface which is exposed to fluid pressure, and let θ be the inclination of the surface ABC to the vertical. Consider any element of area δa at a distance x from the free surface, x being measured along the plane of ABC. The fluid thrust on this element of area

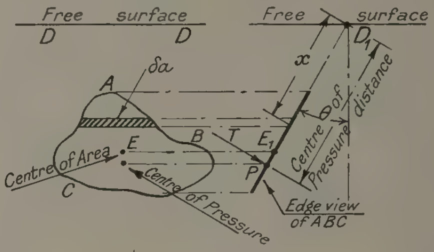


FIG. 148

$$= \delta a \times x \cos\theta \times w \qquad . \quad . \quad . \quad (1)$$

The moment of this force about the line DD

$$= \delta a \times x \cos\theta \times w \times x \qquad . \quad . \quad (2)$$

and the sum of all such elementary moments may be written

$$\Sigma\, \delta a \,.\, x^2 w \,.\, \cos\theta \qquad . \quad . \quad . \quad . \quad (3)$$

= whole moment of fluid thrust about DD.

Again, the whole fluid thrust on the surface $ABC = T =$ area $ABC \times E_1D_1 \cos\theta \times w$, (acting at P, the centre of pressure) and the moment of this force about DD

$$= \text{area } ABC \times E_1D_1 \cos\theta \times w \times PD_1 \qquad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

We can now equate (3) = (4),

$$\Sigma\, \delta a \,.\, x^2 \times w \cos\theta$$

$$= \text{area } ABC \times E_1D_1 \cos\theta \times w \times PD_1 \qquad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Now, $\Sigma\, \delta a \,.\, x^2 =$ the second moment of area of ABC about DD, which is usually written I_{DD}; hence by cancelling and substitution in (5) we have

$$PD_1 = \frac{I_{DD}}{\text{area of immersed surface} \times E_1D_1} \qquad (6)$$

EXAMPLE. The section of a masonry dam used to retain the water of a reservoir is shown in Fig. 149. Determine the water thrusts T_1, T_2, and T_3 per foot length of wall, and the positions of the centres of pressure P_1, P_2, and

P_3 in each case. $FB = 15$ ft., $BC = 15$ ft., $CD = 20$ ft.

By calculation we find $BG = 15$ ft. sec. 15° = 15·529 ft.

$$FK = (GB + BC) \cos 15° = 29{\cdot}488 \text{ ft.}$$

$$CH = FK \sec. 30° = 30{\cdot}05 \text{ ft.}$$

Area FB

$$T_1 = 15 \times 1 \times \frac{15}{2} \times 62{\cdot}5 = 7031 \text{ lb.}$$

$$FP_1 = \tfrac{2}{3}\, FP = 10 \text{ ft.}$$

Area BC

$$T_2 = 15 \times 1 \times 22{\cdot}244 \times 62{\cdot}5 = 20{,}854 \text{ lb.}$$

$$P_2G = \frac{I_G}{\text{area of } BC \times NG}$$

$$P_2G = \frac{[\frac{2}{3} \times 1 \times (7{\cdot}5)^3] + [15 \times 1 \times (23{\cdot}029)^2]}{15 \times 23{\cdot}029}$$

or, $P_2G = 23{\cdot}85$ ft.

Area CD

$$T_3 = 20 \times 1 \times 38{\cdot}148 \times 62{\cdot}5 = 47{,}685 \text{ lb.}$$

$$HP_3 = \frac{I_H}{\text{area } CD \times HQ}$$

$$HP_3 = \frac{[\frac{2}{3} \times 1 \times (10)^3] + (20 \times (44{\cdot}05)^2}{20 \times 44{\cdot}05}$$

or, $HP_3 = 44{\cdot}806$ ft.

FLUIDS IN MOTION

We may consider the flow of a perfect fluid along a channel as consisting of a procession of particles, every one of which follows exactly the path of its predecessor and takes the place of the preceding particle under identical conditions of velocity, pressure, and density. Such motion is termed *steady motion*, and the path followed by any one particle with its stream of successors is termed a *stream line*.

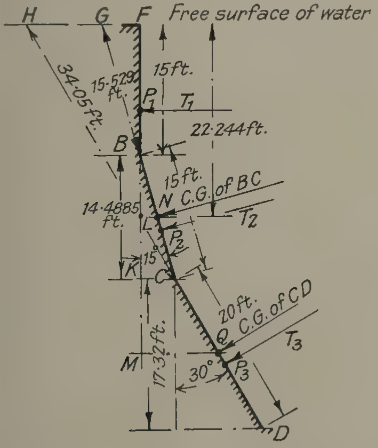


FIG. 149

Steam line motion is a condition of flow which requires certain conditions to be complied with, viz. (1) the fluid is a perfect fluid so that, at any point, the pressure is the same in all directions, (2) an absence of all frictional resistances, (3) an incompressible fluid, and (4) an invariable velocity of any one point in the line of flow.

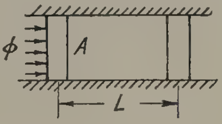


FIG. 150

BERNOULLI'S THEOREM

This theorem is of the greatest importance since it renders invaluable assistance in the solution of problems of fluid flow, as well as having important practical applications. This theorem states that, on the assumptions involved in stream line flow, the *sum of the energies*, taken on a cross-section to the line of flow, *is constant*. Now the energy stored in unit mass of water may be in either or all of three forms, viz. *pressure*, *potential*, and *kinetic*, and *Bernoulli's theorem* can be expressed as

$$\frac{p}{w} + h + \frac{v^2}{2g} = \text{constant.}$$

Where, employing the usual units,

p = pressure in lb. per sq. ft.
w = lb. per cub. ft. of water.
h = height in ft. above datum level.
v = velocity of water in ft. per sec.
g = 32·2 ft.-sec.^{2}.

The work available per lb. of fluid due to the *pressure head* can be determined as follows. Imagine a frictionless piston of cross-sectional area A filling a cylinder, Fig. 150, and acted upon by a fluid at a pressure p; let the stroke of the piston $= L$, so that the volume displaced by the piston $= A \times L =$ volume occupied by 1 lb. of the fluid $= \frac{1}{w}$ cub. ft. Then, the work done by the fluid upon the piston

$$= p \times A \times L$$

$$= p \times \frac{1}{w} \qquad \left(\text{since } A \times L = \frac{1}{w}\right) \qquad (1)$$

and this quantity $\frac{p}{w}$ is the work which 1 lb. of the fluid is enabled to do because it is under a pressure of p lb. per sq. ft.

The store of energy due to the *position* of 1 lb. of fluid is equal to that which it can perform by changing its position

$$h \text{ ft.} = 1 \text{ lb.} \times h \text{ ft.} = h \text{ ft.-lb.} \qquad (2)$$

Conversely, if 1 lb. of the fluid is raised h ft., then an expenditure of h ft.-lb. of work becomes necessary.

When a body is in motion its store of *kinetic energy* $= \frac{Wv^2}{2g}$ (see Lesson XX), or, if $W = 1$ lb., then the work available due to the velocity of the water

$$= \frac{v^2}{2g} \qquad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The sum of (1), (2), and (3) then is constant when all losses are excluded, and we have

$$\frac{p}{w} + h + \frac{v^2}{2g} = \text{constant.}$$

MATHEMATICS FOR ENGINEERS

(*Continued from page* 1308)

Also, $\quad \frac{dy}{d\theta} = 1 + \cos\theta = 1 + 0{\cdot}866 = 1{\cdot}866$

and since $\delta y = \frac{dy}{d\theta}\,\delta\theta$

$$0{\cdot}0236 = 1{\cdot}866\,\delta\theta$$

whence $\quad \delta\theta = \frac{0{\cdot}0236}{1{\cdot}866} = 0{\cdot}0126,$

so $\quad \theta = 0{\cdot}5236 + 0{\cdot}0126 = 0{\cdot}5362$ radians

$$= 30^\circ\,43'.$$

EXERCISE No. 41

1. Differentiate with respect to x—
$\sin 2x$; $\cos 3x$; $a \sin nx$; $4 \sin 5x - 8 \cos 5x$; $3 \sin \frac{1}{2}x + 5 \cos \frac{1}{4}x$.

2. If $s = 6 \cos 10t + \frac{1}{2}(1 - \cos 20t)$ find formulae for $\frac{ds}{dt}$ and $\frac{d^2s}{dt^2}$.

3. The crank of an engine is 6 in. long, and the connecting rod is 27 in. long. Find the piston speed and acceleration at 45°, 90°, and 135° from inner dead centre, the speed being 120 r.p.m.

4. Find the angle at which the speed of the piston in (3) above is a maximum, and the maximum speed.

5. Find the value of θ which makes $4 \cos\theta + 3 \sin\theta$ a maximum, and show that the maximum value of the expression is 5.

6. CD is a chord parallel to the bounding diameter AB of a semicircle, which divides the area into two equal parts. Show that, O being the centre, and $\angle COA$ being denoted by θ, $2\theta + \sin 2\theta = \frac{1}{2}\pi$, and find the value of θ.

7. Starting from the approximation $x = \frac{1}{4}\pi$, to the root of the equation $x = \cos x$, find a better approximation.

8. Show that $\tan A - \tan B = \frac{\sin(A - B)}{\cos A \cos B}$ and by putting $A = \theta + \delta\theta$, and $B = \theta$, deduce that the differential coefficient of $\tan\theta$ is $\sec^2\theta$.

9. Find the values of P and Q if $y = P \sin x + Q \cos x$ makes $\frac{dy}{dx} + 3y = 5 \sin x$.

ANSWERS TO EXERCISE No. 40

1. 201,000 cub. ft.
2. 100·5.
3. 245 cub. in.
4. 1,131 cub. in.
5. 345 lb.-ft.^{2}.
7. $\bar{x} = 2{\cdot}4$, $\bar{y} = 1{\cdot}5$.

POWER TRANSMISSION BY CHAINS

By H. M. Andrew, Wh.Ex.

LESSON I

CHARACTERISTICS OF CHAIN DRIVE

Chain, as a driving medium, has a much wider field of application than is generally realized. Many engineers still think of power drive in terms of belts, ropes, and gearing, but in recent years driving chain has been coming more and more into prominence, and is now beginning to take its proper place as an intermediate form of transmission that combines the positive drive of gears with some measure of flexibility—not enough to allow of any slip; not enough to appreciably affect the positiveness of the drive; but sufficient to absorb a shock or jar in the drive more slowly than gears.

In the same space occupied by a pair of gear wheels it is generally possible to install a comparatively fine pitch chain drive, with chain wheels nearly touching one another. Such a drive, at fairly high speed, has an exceptionally high efficiency, in the order of 99 per cent or over, and this efficiency would probably fall no lower than 98 per cent throughout the life of the drive. The percentage pitch extension by wear, however, varies almost inversely as the centre distance of a chain drive and, with such a short centre distance, the useful life of the chain would be short in spite of its high mechanical efficiency, unless a third adjusting wheel was installed, or one shaft was movable.

At the other extreme—in the same space occupied by a long horizontal belt or rope drive, a chain drive could be installed on large wheels that would also have a fairly high mechanical efficiency. But here again, although the individual pitch extension or wear would be small, the long life of the chain could only be taken advantage of by periodic and regular adjustment, as even a small pitch extension would give enough total extension in a very long drive to cause the chain to lose its positive grip on the driver wheel. A chain is similar in action to a belt with very little internal friction and a positive grip. This small internal friction, while very efficient as regards power loss, has no "damping" effect to prevent flapping; and a long chain drive, unless maintained in close adjustment, would in time tend to ride the teeth of the driver wheel. Another objection to long horizontal chain drives is the catenary pull caused by the weight of the chain between the wheels, particularly when close adjustment is maintained.

Like belts and ropes, chain has centrifugal tension, possibility of flapping between wheels, and the necessity for periodic adjustment.

Unlike these, it is a positive drive and has very little internal friction.

While chain drive encroaches in this manner on the particular province of gears at one end of the scale and belts or ropes at the other extreme, its natural field is between the two, where the drive is from, say, 18 in. to 15 ft. centre distance for horizontal drives, up to, say, 30 ft. (with close adjustment) for vertical drives, and within this field it is an excellent method of transmission of very high and maintained efficiency—if suitable conditions are provided.

To explain what is meant by "suitable conditions" it will be necessary to consider the method of use of driving chains, leaving the design and construction of these chains, to be explained fully in the next lesson.

The Bearing Pressure at which chains work is considerably higher than it would be practicable to employ with machine bearings, partly because the permissible rate of wear is higher, but mainly because the average time rate of movement in the bearing is very slow and the pressure fluctuates steadily between zero and maximum.

Manufacturers base their catalogue ratings on a bearing pressure of about 2,000 lb. per sq. in., which, if used to the full, gives an average factor of safety of about 20, a little less in the large pitch roller chains and a little more in small pitch roller or inverted tooth chains.

The Load Pull in the chain is, of course, the work done in foot-pounds divided by the linear speed of the chain in feet per minute. For a given revolution per minute the pull in the chain drive due to this varies inversely as the pitch diameter of the driving chain wheel.

With a bearing pressure of 2,000 lb. per sq. in., the speed at which highest mechanical efficiency is obtained is about 2,500 ft. per minute for bush roller, and 1,700 ft. per minute for inverted tooth chain. The efficiency falls off above this speed and, at, say, twice this speed, is falling off rapidly. Below this speed, on the other hand, the efficiency only falls off slowly so that, while the best results are obtained at about this speed, it does not affect the mechanical efficiency much to work below it.

While we should never exceed this speed there is good argument for keeping as near to it as is practicable, because that gives us large wheels, and large wheels tend to freedom from noise and polygonal effect, provided always that we reduce the pitch of the chain at the same time that we increase the wheel diameter.

The Chain Pitches form a polygon around the chain wheel, and if the wheel were turned very slowly the chain line entering the wheel would rise and fall between the flat side and the point of the polygon. But in all industrial drives the speed is amply sufficient to cause the chain line to assume a permanent intermediate position, leaving individual links to be lifted slightly to the point of the polygon. In effect, the chain falls slightly by gravity and is met by the vertical component of the wheel velocity, causing a slight impact. This impact, which is the cause of noise in a chain drive, is, of course, immensely accentuated by any flapping of the chain between the wheels.

In the case of roller chain, before any wear has occurred, there is theoretically a direct radial impact of the roller in the tooth gap, but in practice the roller probably glances down the tooth side a little and certainly does so after wear has occurred. If there is little or no oil on the chain the roller may rattle on its bush. In the case of inverted tooth chain, the link face slides down the side of the wheel tooth and there is nothing to cause any rattle. Hence the term " noiseless " or " silent " chain.

Bush roller chain can, however, also be very quiet when given radial lubrication to cushion the rollers on to their seat.

Wear. As wear occurs in the bearings of a chain the chain pitches lengthen very slightly. The case-hardened bearings of good quality driving chains are hardened to a depth sufficient to give about 3 per cent pitch extension before the hardened surface is worn through. As the pitch lengthens, the centrifugal tension and the load tension cause the chain to ride out on the teeth of the wheels and assume a larger pitch circle proportionate to the increase in pitch by wear.

Chain wheels have sufficient height of tooth, in wheels up to about 80 teeth, to fully engage the chain for the whole of its 3 per cent wear or life. But with 120 teeth the permissible wear is only 2 per cent, and with 150 teeth, say, $1\frac{3}{4}$ per cent, before the chain begins to tend to ride over the tops of the teeth. This tends to set a limit to the number of teeth that should be used, but it must be kept in mind that large numbers of teeth mean much less *rate* of wear and, therefore, 2 per cent wear on a large number of teeth may often mean a longer life than 3 per cent wear on a small number of teeth.

As 50 to 60 teeth is as much as one needs in order to secure excellent efficiency, the above limit only applies in cases where large speed reduction ratio is called for.

As regards wear, prevention is better than cure, and the preventive medicine is pure mineral oil applied more or less copiously, the amount varying, say, as the *square* of the linear chain speed. Chain drives, like other mechanisms, cannot be expected to move without oil. Industrial drives installed under reasonably good conditions, with good lubrication, generally give a life of from seven to fifteen years.

Good lubrication is of very special importance. For high speed drives the oil should be squirted on to the *inside* face of the chain just where it is entering a wheel. It is a good plan to recess the inside face of the wheel rim and bore several small holes from this recess through to the space between the wheel teeth ; oil poured in this recess will then be thrown out radially through the holes on to the inside face of the chain. When wear *has* occurred, the cure is adjustment ; preferably by moving one shaft away from the other, as can be done with a motor on slide rails. If this is not permissible, a pitch can be taken out of the chain or, if this is not a fine enough method, a jockey wheel can be used. The jockey wheel, of course, introduces further wear points in flexing the chain, but if placed on the slack side of a moderate speed drive, this effect is very small.

STEAM TURBINES

By W. J. Kearton, M.Eng. A.M.I.Mech.E., A.M.Inst.N.A.

LESSON X

THE RADIAL-FLOW REACTION TURBINE

Most modern turbines operate on the axial-flow principle, i.e. the general direction of steam flow through the turbine is axial. In some of the early turbines built by Parsons, the flow was radial. The expansion of the steam was carried out in a certain number of stages, and each stage comprised a number of circular concentric rings of blades attached to a disc on the shaft and running in spaces between similar rings of blades attached to the stator or casing.

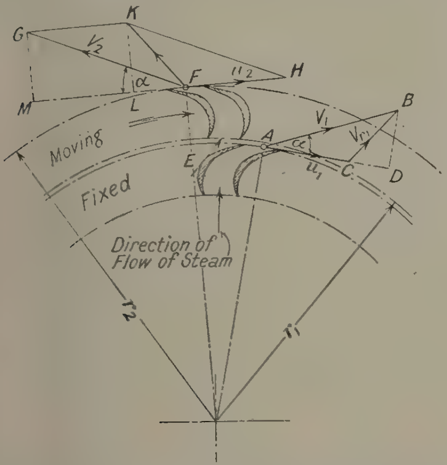


Fig. 43. Velocity Diagrams for Single-motion Radial-flow Turbine

Principle of Action of Single-motion Radial-flow Reaction Turbine. Fig. 43 shows diagrammatically a *single-motion* radial-flow turbine. In general, there would be several stages in series, but our sketch only shows one elementary stage comprising an inner *fixed* ring and an outer *moving* ring of blades. In flowing through the inner ring of blades, the steam is partially expanded and issues with a velocity AB relative to the fixed blades. If AC be drawn to represent the peripheral velocity of the moving blades, then the relative velocity of flow into the moving blades will be given by CB, and is represented in the sketch by the arrow E.

In passing through the moving ring of blades, the velocity of the steam is increased from CB to FG owing to further expansion. If FH be drawn to represent to the scale of velocities the peripheral velocity at r_2, then the parallelogram of velocities may be drawn, giving the final absolute velocity of the steam as FK.

Now, if a body of mass M lb. be moving in a circular path and a force of P poundals acts on it, then it will produce an acceleration of circumferential velocity equal to

$$f = \frac{P}{M} \text{ ft. per sec.}^2$$

and in t sec., the change of velocity will be

$$V = ft = \frac{Pt}{M}$$

$$\text{or } P = \frac{MV}{t} \text{ and } P = \frac{MV}{gt} \text{ lb. weight}$$

Multiplying by the radius r of the path,

$$\text{Torque } T = \frac{(MV)r}{gt} \text{ ft.-lb.}$$

i.e. *Torque* $= \frac{1}{g}$ (*Change of Moment of Momentum per sec.*).

Referring to Fig. 43, the initial moment of momentum is $AD \,.\, r_1$ per lb. of steam, and the final moment of momentum is $-FL \,.\, r_2$ per lb. of steam. Note that since FL is in the reverse direction to AD, we prefix a minus sign before the quantity $FL \,.\, r_2$.

Hence, if w = weight of steam flowing through such a turbine per sec., then

$$\text{Torque} = \frac{w}{g}\left\{AD \,.\, r_1 - (-FL \,.\, r_2)\right\}$$

$$= \frac{w}{g}\left(AD \,.\, r_1 + FL \,.\, r_2\right)$$

and if ω = angular velocity of the shaft in radians per sec.

Work done per sec.

$$= \omega \,.\, T \text{ ft.-lb.}$$

$$= \frac{w}{g} \,.\, \omega \,.\, (AD \,.\, r_1 + FL \,.\, r_2)$$

$$= \frac{w}{g}\,(AD \,.\, u_1 + FL \,.\, u_2)$$

But $AD = V_1 \cos a$ and $FL = FM - ML$

$$= V_2 \,.\, \cos a - u_2$$

$\therefore$ Work done

$$= \frac{w}{g}\,\{u_1 \,.\, V_1 \cos a + u_2\,(V_2 \,.\, \cos a - u_2)\}$$

If the radial depth of the blades is small compared with r_1 and r_2, then, approximately, u_1 is equal to u_2, and with "half-degree reaction," V_2 equals V_1.

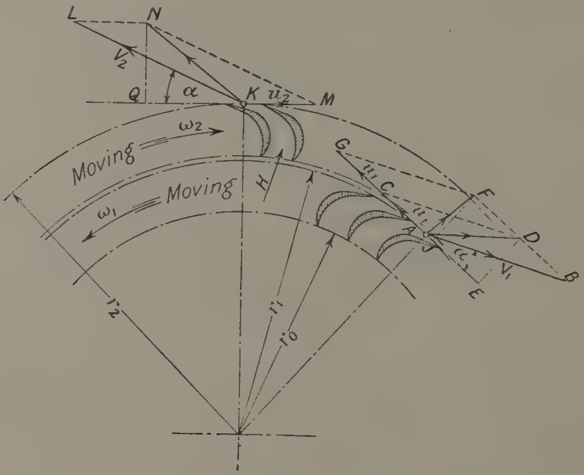


Fig. 44. Velocity Diagrams for Double-motion Radial-flow Turbine

Further, writing $\rho = \frac{u_1}{V_1} = \frac{u_2}{V_2}$, we have,

Work done by w lb. of steam

$$= \frac{w \,.\, V_1^2}{g}\,(2\rho \,.\, \cos a - \rho^2).$$

Now, the energy given to the steam by expansion in the blades is $\frac{wV_1^2}{2g}$ ft.-lb. (approx.) in each row, i.e. the total energy supplied is $\frac{w \,.\, V_1^2}{g}$

$$\therefore \text{ Efficiency} = \frac{w \,.\, V_1^2}{g}\,(2\rho \,.\, \cos a - \rho^2) \times \frac{g}{w \,.\, V_1^2}$$

$$,, \quad = 2\rho \,.\, \cos a - \rho^2$$

For maximum efficiency, $\frac{d\eta}{d\rho} = 0$

$$\therefore 2 \cos a - 2\rho = 0$$

$$\text{or } \rho = \cos a$$

Thus the ratio of peripheral velocity to steam exit velocity for maximum efficiency is exactly the same for the radial-flow as for the axial-flow turbine.

Principle of the Double-motion Radial-flow Reaction Turbine. Fig. 44 represents, in diagrammatic form, two adjacent rings of blades in a *double-motion* turbine. *Both* rings rotate in this instance, but in *opposite directions*.

In the inner ring of blades, the steam is partially expanded and flows out with a velocity AB relative to the blades. The steam has, however, a peripheral velocity AC in common with the blades themselves. Drawing the velocity parallelogram $ABCD$, we obtain AD, which is the absolute steam velocity.

If we assume that the steam possesses no tangential velocity at inflow to the ring of blades in question, then the work done on the inner ring of blades per w lb. of steam

$$= \frac{w}{g} \, . \, u_1 \, . \, AE = \frac{w}{g} \, . \, u_1(V_1 \cos \alpha - u_1)$$

In order to obtain the velocity of the steam relative to the outer blades at inlet, we observe that the relative velocity of the two blade rings is $2u_1$ (assuming the angular velocities ω_1 and ω_2 to be equal), and we, therefore, draw AG equal to $2u_1$ and complete the parallelogram $ABFG$. AF is then the velocity of steam relative to the second ring of blades. The arrow H drawn in the same relative direction as AF shows the direction of inflow.

In the outer ring of blades, the steam acquires a velocity V_2 due to expansion, and this, combined with the peripheral velocity $KM = u_2$, results in a final absolute velocity KN.

The change of moment of momentum in the outer blade ring is

$$w(KQ \, . \, r_2 + AE \, . \, r_1) \, \frac{\text{lb.-ft.}^2}{\text{sec.}} \text{ per } w \text{ lb. of steam}$$

$\therefore$ Work done on the outer ring of blades per w lb. of steam

$$= \frac{w}{g} \, (u_2 \, . \, KQ + u_1 \, . \, AE) \text{ ft.-lb.}$$

$$= \frac{w}{g} \, \{u_2(V_2 \, . \cos \alpha - u_2) + u_1(V_1 \, . \cos \alpha - u_1)\}$$

If we again assume that the radial depth of each blade ring is small compared with either r_1 or r_2, and also that the degree of expansion is the same in both blade rings, then $u_1 = u_2$ and $V_1 = V_2$.

$\therefore$ Work done in the outer ring of blades

$$= \frac{w}{g} \, \{2u_1(V_1 \, . \cos \alpha - u_1)\}$$

and the total work done in both rings of blades

$$= \frac{w}{g} \, \{3u_1(V_1 \cos \alpha - u_1)\}$$

$$= \frac{3w \, . \, V_1^2}{g} \, (\rho \, . \cos \alpha - \rho^2)$$

where $\rho = \dfrac{u_1}{V_1}$

The energy supplied in the two rings is *approximately* $2 \times \dfrac{w \, . \, V_1^2}{2g}$

and therefore the efficiency

$$= \eta = \frac{3w \, . \, V_1^2}{g} \, (\rho \, . \cos \alpha - \rho^2) \times \frac{g}{wV_1^2}$$

$$= 3(\rho \, . \cos \alpha - \rho^2)$$

This is obviously a maximum when $\rho = \dfrac{\cos \alpha}{2}$

and the maximum value of η is

$$\eta_{max} = \frac{3 \, . \cos^2 \alpha}{4}$$

If the two blade rings in question were in the midst of other blade rings, then the conditions at inlet for the inner ring would be analagous to those for the outer ring, and the work done in the inner ring per w lb. of steam would be

$$\frac{w}{g} \, \{u_1(V_1 \cos \alpha - u_1) + u_0(V_0 \, . \cos \alpha - u_0)\}$$

and, again assuming that $u_0 = u_1 = u_2$; and $V_0 = V_1 = V_2$, then the total work done in the two rings

$$= \frac{4w}{g} \, . \, u_1(V_1 \cos \alpha - u_1)$$

$$= \frac{4w \, . \, V_1^2}{g} \, (\rho \, . \cos \alpha - \rho^2)$$

$$\therefore \; \eta = 4(\rho \, . \cos \alpha - \rho^2)$$

The efficiency is again a maximum when $\rho = \dfrac{u}{V} = \dfrac{\cos \alpha}{2}$, and its maximum value

$$\eta_{max} = \underline{\cos^2 \alpha}$$

Importance of Double Motion. For a given peripheral velocity u, the steam velocity is about twice as great as in the single-motion reaction turbine, and hence the heat drop utilized is about four times as great. In other words, for a given total heat drop, the number of stages required is reduced to one-quarter.

RECIPROCATING STEAM-ENGINES

By Engineer Lieut.-Commander T. Allen, M.Eng., R.N. (s.r.)

LESSON XI

ENGINES DESIGNED FOR USE WITH EXHAUST STEAM

(contd.)

Steam Extraction Engines. The difficulty of balancing the demands for power and process steam necessitates, in most instances, the employment of a modified type of engine known as the steam extraction engine. In the horizontal form this is a compound engine arranged as shown diagrammatically in Fig. 33, and in Figs. 34 and 35. Steam from the boiler is led to the high pressure cylinder *A*, Fig. 34 (facing page 1305), through pipes *B*, and generates power in this cylinder during expansion to the required process steam pressure. The exhaust steam from the high pressure cylinder enters the receiver *C*, Fig. 33, and the proportion required for process purposes is drawn from this receiver through branch *D*. The surplus passes through pipes *E* to the low pressure cylinder *F*, where it does further work during expansion to the pressure in the condenser *G* or G_1. The reciprocating air pump of the engine shown in Fig. 34 is driven from an overhung crank on the engine crankshaft, and the jet condenser bolted directly to the exhaust branch of the low pressure cylinder. In Fig. 33, both ends of the crankshaft accommodate half couplings for the direct transmission of power—in addition to rope drive from a pulley alongside the fly-wheel. In this case, therefore, the air pump is of the steam operated ejector type, and the jet condenser is arranged independently in the basement with an electrically driven centrifugal water extraction pump.

Governing System. The usual requirements of constant speed and process steam pressure are met by regulating the cut-offs of the high pressure cylinder through the action of a speed governor, and those of the low pressure cylinder by means of a pressure governor (see Fig. 35). If the load remains steady and the demand for process steam increases, the pressure in pipe *D* tends to fall, and the pressure governor reduces the cut-off of the low pressure valves, thus curtailing the supply to the low pressure cylinder and causing a reduction in output and a tendency to reduced speed. The speed governor thereupon comes into operation, increasing the cut-off of the high pressure valves and restoring the receiver pressure and combined output to normal. The reverse takes place if the demand for process steam is reduced, and similar regulation occurs if the process demand remains constant and the load changes.

The dual governing system works quickly and effectively in practice, and the usual speed uniformity may be obtained with practically constant pressure of extracted steam. It will be noted that the engine provides automatically for variation in load and process steam demand between wide limits. At times of maximum extraction, the cut-off of the low pressure cylinder may be reduced to such a limit as serves only to prevent the piston working " dry," and the high pressure cylinder develops practically all the load, so that the engine works with an economy approaching that of the back pressure engine. When the demand for process steam is nil, the load is shared between the two cylinders, and the engine works with an economy approximating to that of an ordinary compound condensing engine. As with the back pressure engine, the steam consumption is reduced by lowering the extracted steam pressure, and by adopting as high an initial superheat as possible. Also, the heat loss to the condenser is reduced and the thermal efficiency improved, by increasing the amount of steam extracted from the intermediate receiver. Owing to the great variation of power and piston load distribution, the engine is usually constructed in the tandem form, although the cross-compound arrangement may be adopted in suitable cases.

Automatic reducing valves—opened by the pressure governor when the cut-off in the low pressure cylinder has been reduced to a minimum—may be fitted to supply any deficiency of process steam, or to by-pass steam from the boilers when the engine is stopped.

Description of Steam Extraction Engines. Fig. 35 is reproduced from a photograph of a horizontal steam extraction engine built for a normal load of 650 i.h.p. at 160 r.p.m., and extraction steam quantities up to 11,000 lb.-hr. at 10–15 lb. gauge pressure. This engine is direct-coupled to a 200 kW. D.C. generator by means of a coupling beyond the outer bearing shown in the photograph, and the remainder of the power is transmitted through ropes.

The general constructional features are similar to those of the back pressure engine—with the addition of the low pressure cylinder—and the cylinder ratios require careful selection in each particular case. As a rule, the high pressure cylinder is larger and low pressure cylinder smaller than those of ordinary compound engines, and for high extraction rates the cylinder diameters may approach equality. Where low pressure cylinders are of the Unaflow

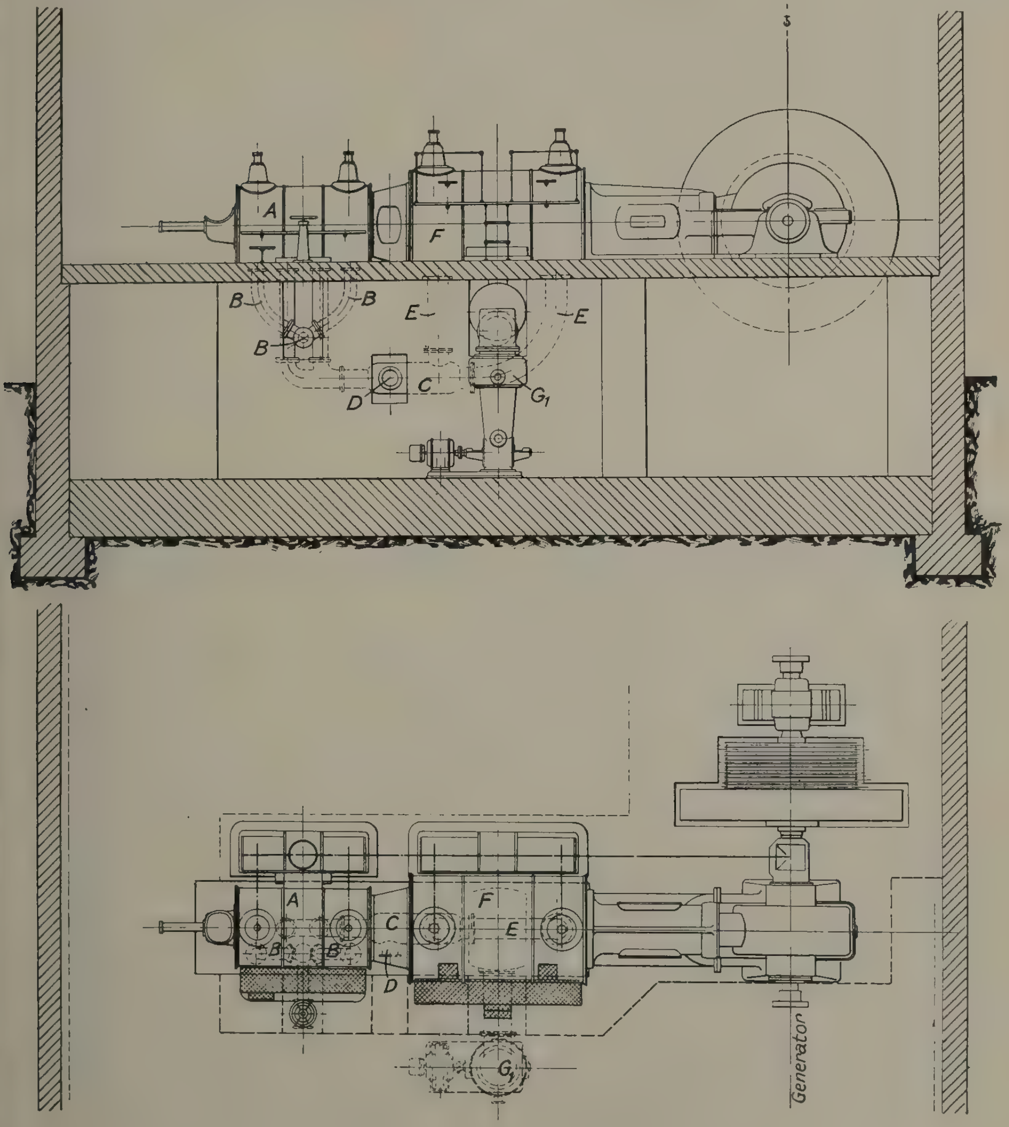


Fig. 33. Diagrammatic Arrangement of Steam Extraction Engine

type, means must be provided to avoid excessive compression at starting, or at times of reduced vacuum as described for normal Unaflow engines. The engines are, at present, usually designed for steam pressures from 150–250 lb. per sq. in., and receiver pressures up to a normal maximum of about 70 lb. per sq. in., although this pressure should be kept as low as possible for reasons already stated. Vertical high speed engines may also be arranged for the extraction of steam from a special low pressure receiver. An automatic valve is fitted to control the supply of steam to the low pressure cylinder; all steam not required for process purposes passing through this cylinder and thence to the condenser.

Example of Savings. The approximate saving due to the installation of a steam extraction engine—rather than a pure condensing engine for power, with separate supply of the process steam—may again be illustrated by a simple example—

Initial steam	190 lb. per sq. in. gauge pressure, superheated to 550° F.
Normal process steam demand	10,000 lb. per hr. at 40 lb. per sq. in. gauge pressure.

Normal load demand 600 i.h.p.

	lb. per hr.
Total live steam required by horizontal steam extraction engine for power and process steam requirements	= 12,400
Live steam required, say, by condensing Unaflow engine for power requirements only . . .	= 6,300
Add live steam equivalent of 10,000 lb. per hr. process steam required	= 9,100
Total live steam required for condensing scheme	= 15,400

Saving in favour of steam extraction engine = 15,400 – 12,400 = 3,000 lb. per hr. or about 19·5 per cent over the condensing scheme.

The best case has purposely not been chosen in this example, inasmuch as the extraction pressure is somewhat high and the extracted quantity comparatively low. The steam extraction engine, working at the load specified, could yield process steam quantities up to about 15,000 lb. per hr. without make-up through the auxiliary live steam valve. The economy of the engine would increase with an increase in the extracted steam quantity, and better results would also be obtained by lowering the

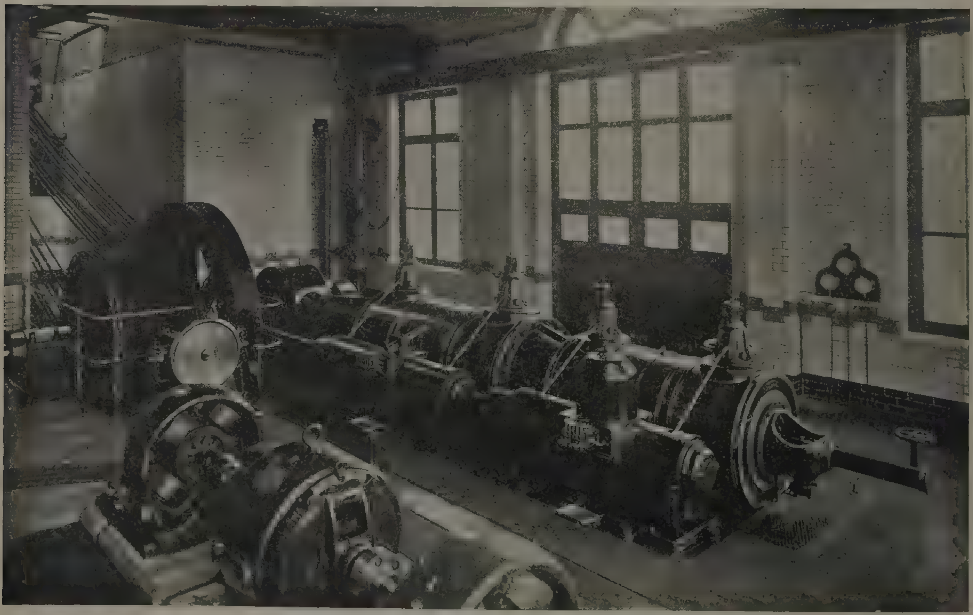


Fig. 35. 650 I.H.P. "Musgrave" Steam Extraction Engine

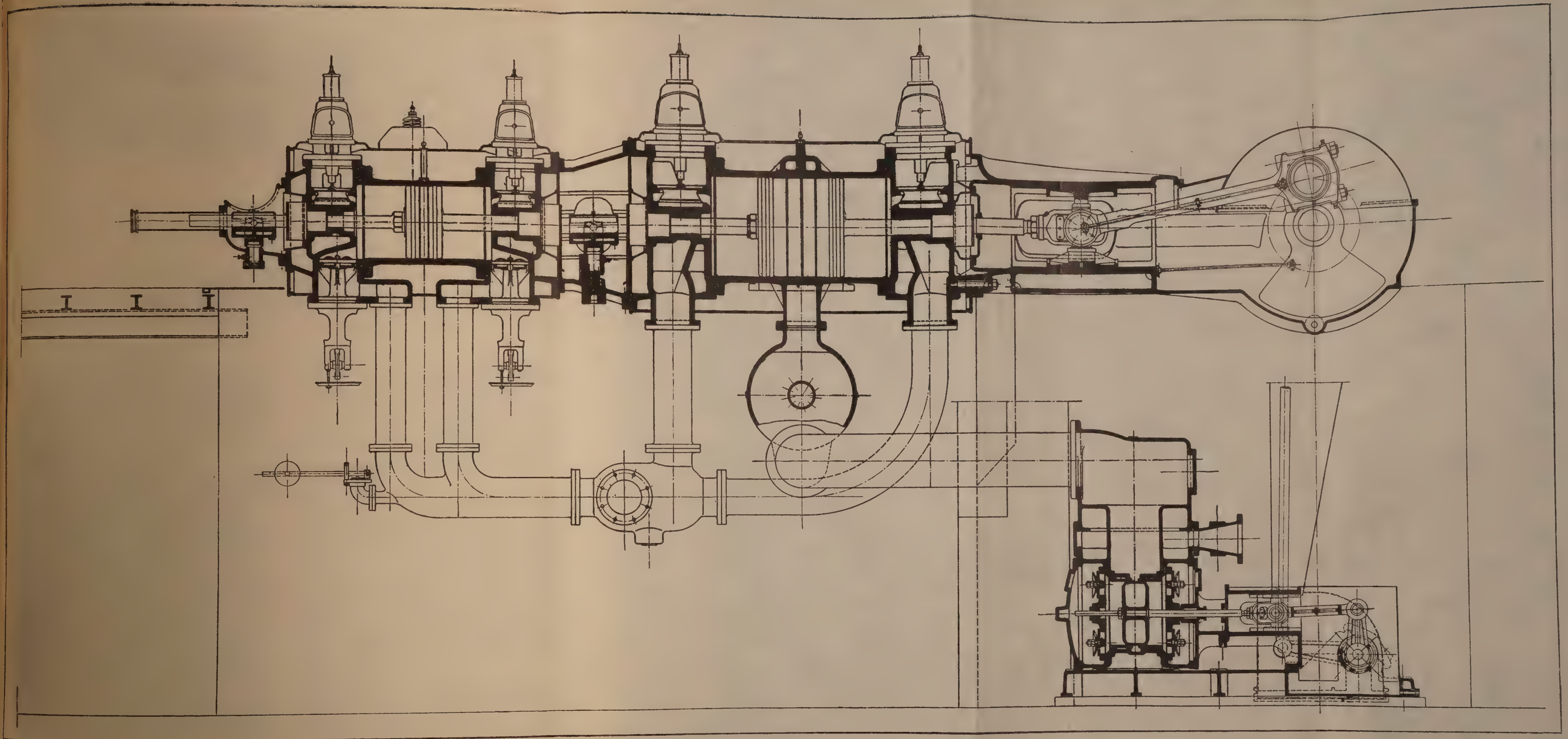
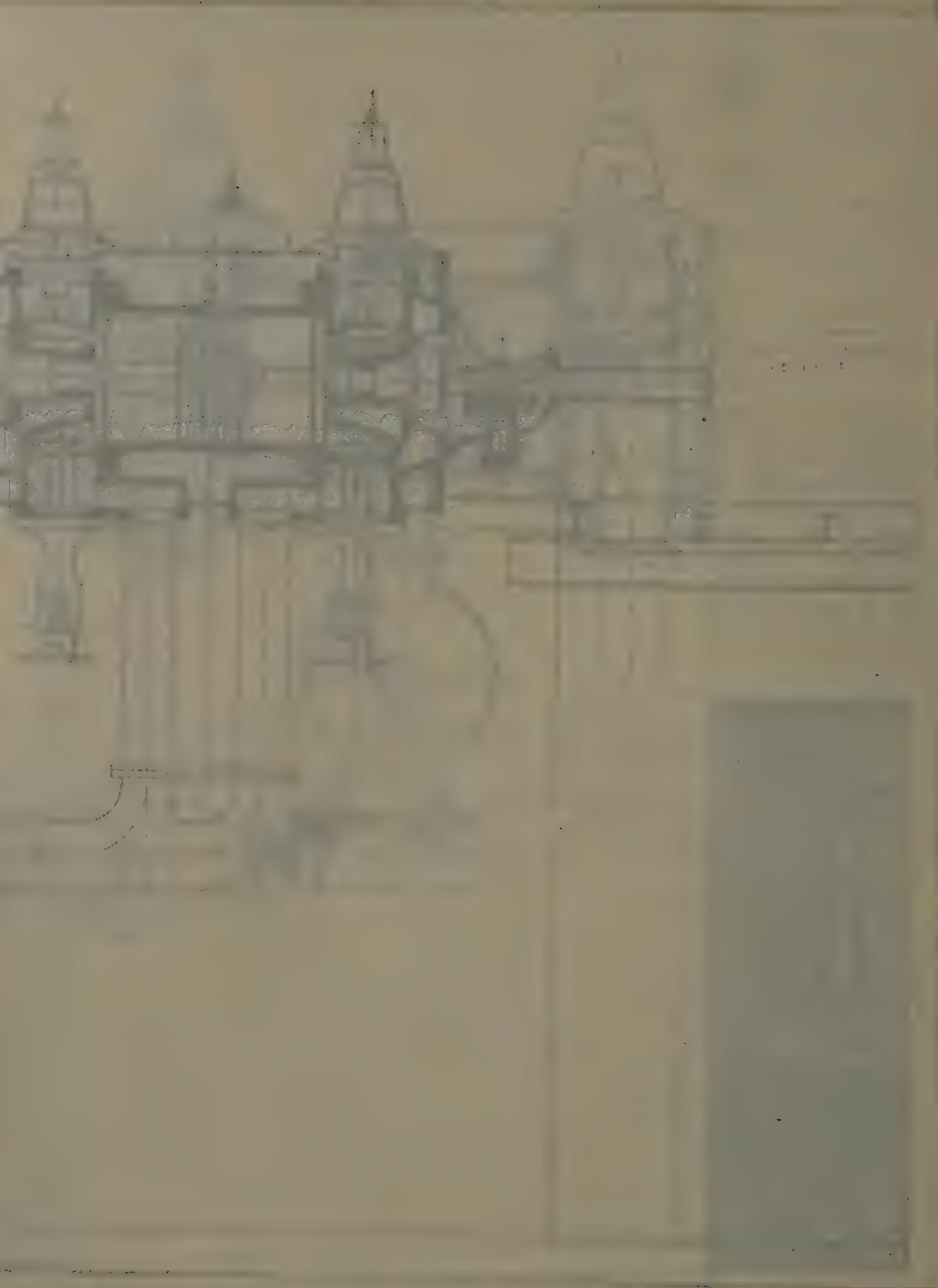


Fig. 36. Longitudinal Section through Tandem Steam Extraction Engine

H.P. Cylinder of the " Counterflow " type
L.P. Cylinder of the " Unaflow " type



extraction pressure. In such cases the comparison is even more favourable, and the saving in fuel is approximately proportional to the steam saving. A further advantage accrues owing to the reduced quantity of condensing water required.

Conclusion. The overall economy of engines working in conjunction with plant which utilizes the heat content of exhaust steam is, in many cases, greatly superior to that of the most modern power station. It is not surprising, therefore, that the system of steam extraction has been widely adopted on the Continent where coal is expensive, and, to a smaller extent, in this country. Even to day, however, a wide field exists for the application of steam extraction engines, and—apart from the economies consequent upon reduced manufacturing costs—their extended use would be accompanied by a very large national saving of fuel.

(Conclusion)

DIESEL ENGINES

CONSTRUCTION, OPERATION, AND MAINTENANCE

By A. Orton, A.M.I.Mech.E.

LESSON XI

Operation and Maintenance. We shall now endeavour to give some hints about the special points that require attention in the operation and maintenance of Diesel engines as distinct from the ordinary procedure and practice in connection with internal combustion engines generally.

Regulation of the Blast Air. In a normal Diesel engine it is necessary, in order to obtain the most efficient working, to regulate the blast pressure to suit the load on the engine. The reason for this may be explained as follows—

The fuel cam and valve is necessarily designed to admit the right quantity of oil when the engine is working at its full capacity and the blast air is at a certain pressure. At lower loads the quantity of oil used is smaller, while the valve opening remains unaltered. The oil in the pulverizer parts increases the resistance to the flow of the blast air through them, and therefore, when the quantity of oil is less the resistance is less and more blast air will pass through into the cylinder under the same blast pressure. This air is, of course, cold and at a high pressure, and enters the cylinder containing very hot air at a much lower pressure. The blast air, therefore, in expanding to the lower pressure and at the same time being raised to a very much higher temperature, takes up considerable heat and, consequently, has the effect of reducing in some degree the temperature of the air in the combustion space.

It follows, therefore, that, with the same blast pressure, more cold air would enter the cylinder at low load than at full-load, and this would cause such a greater cooling effect as to interfere to a noticeable degree with good combustion, causing the phenomenon known as a smoky exhaust.

The simplest way of remedying this is to reduce the blast pressure as the load decreases, thereby reducing the quantity of cold air entering the cylinder. As an indication of the extent of regulation necessary, it may be stated that if an engine works at full-load with a blast pressure of 900 lb. per sq. in., then at no-load the blast pressure would be about 540 lb. per sq. in., and at intermediate loads the pressure would lie proportionately between these two values. In practice too high a blast pressure at no-load gives the exhaust gases a whitish colour, while at heavier loads with too low a blast the gases become blackish.

It follows from the above facts that on a fluctuating load, it is necessary, for the most economical working, to adjust the blast pressure to suit any considerable change in the load on the engine. This, of course, entails more or less constant attention on the part of the attendant and, in order to avoid this, some makers supply an automatic regulating device which controls the blast pressure in accordance with the load, thus ensuring the best conditions of combustion at all times with the maximum of efficiency and cleanness, as the sooting-up of the piston top, cylinder heads, and valves due to incomplete combustion, resulting from an unsuitable blast pressure, is materially reduced, if not entirely avoided.

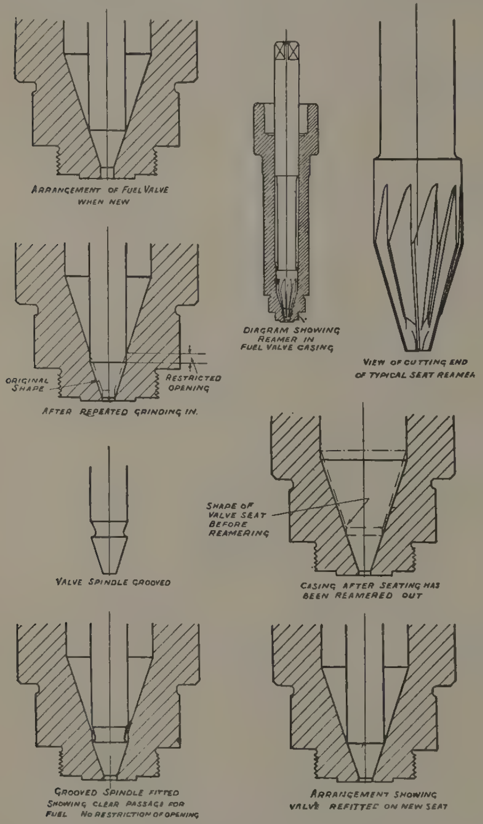

ARRANGEMENT OF FUEL VALVE WHEN NEW
DIAGRAM SHOWING REAMER IN FUEL VALVE CASING
VIEW OF CUTTING END OF TYPICAL SEAT REAMER
ORIGINAL SHAPE
RESTRICTED OPENING
AFTER REPEATED GRINDING IN.
VALVE SPINDLE GROOVED
SHAPE OF VALVE SEAT BEFORE REAMERING
CASING AFTER SEATING HAS BEEN REAMERED OUT
GROOVED SPINDLE FITTED SHOWING CLEAR PASSAGE FOR FUEL NO RESTRICTION OF OPENING
ARRANGEMENT SHOWING VALVE REFITTED ON NEW SEAT

FIG. 41. FUEL VALVE SEAT SHOWING EFFECT OF WEAR AND METHODS OF REPAIR

Valves in Cylinder Cover. It is vital to keep these valves gas and oil tight, and in a free working condition. The fuel valve particularly must be kept free in its packing, if of the packed type, so that there is no danger of sticking open, thereby permitting leakage of fuel and blast air into the cylinder or causing injection at the wrong time. In some designs packing is dispensed with, as a means of avoiding this possible trouble. Regular cleaning of the pulverizer parts is necessary, the frequency depending chiefly upon the nature and quality of the fuel oil used. The needle valve should also be rubbed or ground on its seat fairly regularly. Some oils are thick and sticky by nature or are difficult to burn completely, such as those of strongly asphaltic origin, and this tends to cause pitting of the surfaces of the valves and their seats and, consequently, a lack of complete tightness, which leads to high exhaust temperatures and a dirty exhaust.

It may happen that the frequent grinding of the fuel valve, particularly if a hard abrasive is used, a practice which is not to be recommended, produces a recess in the seat of such a depth as to bring about a material reduction in the effective opening of the valves and, consequently, to cause interference with the passage of the oil and blast air. Such a condition is illustrated in Fig. 41, which also indicates two methods of rectifying it, i.e. by grooving the needle valve or by reamering out the seat by means of a special tool. The main effects of such a condition of the valve and seat are loss of power, difficulty of starting and local overheating.

In long fuel valve casings it is, obviously, rather a difficult matter to discover the true condition of the seat, and to measure the actual depth of any recess there may be in it. Perhaps the neatest method of doing this is that due to Mr. G. E. Windeler,[1] in which a softened wax candle is pressed down firmly into the casing so that it takes the actual form of the seat, which is clearly shown when the candle is withdrawn, after being allowed to harden.

The pulverizer parts need cleaning more or less often, according to the quality of the oil in use, while the actual nozzle or flame-plate seldom requires much attention in the air injection type of engine, owing to the comparatively large size of the hole, or holes, through it, which under ordinary circumstances do not readily become choked.

As regards the packing of the fuel valve spindle several different forms are in use, one of the commonest being made up of metallic lead shavings compounded with graphite. This gives good service, as also do various makes of fibrous material graphitized in a similar way. There is, however, a strong tendency in modern designs to avoid the use of packings altogether, and to rely on good mechanical fits or special mechanical devices to prevent leakage of blast air or fuel oil from the fuel valve casing.

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[1] See *Care and Maintenance of Diesel Engines*, by G. E. Windeler. Paper before The Diesel Engine Users Association, 27th October, 1922, from which the above Fig. 41 is taken.

It is important to attend regularly to the exhaust valves, a good plan being to change these after a certain period of running, replacing them by spare ones. They can then be cleaned and ground in their seats at leisure, and thus be ready for insertion at any moment. The air inlet valve and the starting valve require comparatively little attention, other than very occasional grinding in, as these are not subject to the flow of the hot gases of combustion over their seats. As in the case of the fuel valve, the quality of the fuel oil affects the question of how frequently the exhaust valve requires attention, the use of heavy viscous oils naturally necessitating more attention than if lighter and more fluid ones are used. In all cases, however, it pays to adopt a regular and systematic plan of inspection and cleaning, thus ensuring that these vital parts are kept in a thoroughly efficient working condition.

Main Pistons. As with all other internal combustion engines, periodic inspection of the pistons is advisable. After lengthy running there is a tendency for the sides near the combustion end to become coated with hard carbonized oil, which sometimes gums up the piston rings, preventing their free expansion in the grooves and thus causing leakage and loss of compression. When this occurs the pistons should be withdrawn, cleaned, and the rings freed in their grooves. The chief factor that affects this particular point is the lubrication of the cylinder and piston. Too little oil may lead to a dry piston and seizure, whilst too much may cause a dirty piston head, clogging of the exhaust valves and a bad exhaust.

The piston is made as close a fit in the cylinder as possible, allowing for expansion and adequate side clearance when working at the maximum rated overload. Under ordinary circumstances an engine would run almost indefinitely without piston trouble, but if for any reason, such as the failure of the water supply due to accident or carelessness, or the running of the engine for any considerable time above its rated overload, the piston becomes too hot, a seizure will take place between it and the cylinder walls. This may lead to serious results and it is, therefore, of the utmost importance to avoid such an occurrence. As previously mentioned, large pistons are separately cooled by the internal circulation of water or oil, and it is obvious in such cases that any failure of the supply would quickly bring about a dangerous temperature in the material of the piston.

To avoid piston trouble, therefore, there are three important things to watch ; not to overload the engine unnecessarily, to maintain correct lubrication, and to maintain efficient cooling. In regard to the latter it is not merely sufficient to see that the correct quantity of water is flowing to each cylinder, but it is also necessary to see that the water spaces in the cylinder casing and cylinder cover are kept reasonably clean. A very frequent cause of trouble is the deposition of mud or hard impurities from the water in the water spaces, which brings about local overheating and, perhaps cracking of the metal at some point where it is subject to high working temperatures. One very important yet simple method of reducing deposits of this kind is to arrange that the cooling water can be circulated through the cooling spaces for some time after the engine has stopped running. In this way, boiling of the water in the jackets due to the heat retained in the working parts is avoided, and the consequent deposition of the impurities in the water materially reduced. Where the cooling water is taken from a river or stream liable at times to become muddy, the above-mentioned precautions are particularly important. One of the advantages of the closed circulating water system, that is a system where the same water is used continually and passed through a cooler, is that the chances of deposits forming in the jackets are almost entirely avoided.

OUTLINES OF LOCOMOTIVE ENGINEERING

By A. Morton Bell, O.B.E., M.I.Mech.E., M.I.Loco.E.

LESSON VII

LUBRICATION

The efficient lubrication of the various parts of a locomotive requires very careful attention and study, and is a detail on which time and money can be well spent.

It is manifest that the greater the friction existing between any two bodies, the more will be the power required to move them. The introduction of a lubricating medium between the different surfaces, more especially those of the valves, cylinders, etc., can be secured either by mechanical means or by displacement. For the axle-boxes and journals, capillary, or natural gravitation are usually used, with pipe connections from the foot-plate or other suitable position; forced lubrication by mechanical means for the axle-boxes is in use.

Lubricators of the so-called mechanical order consist of a series of miniature pumps, which force the oil in measured quantities along pipes to the different details requiring lubrication, whilst those depending on the hydrostatic or displacement of the lubricant by water, have a chamber under pressure into which condensed steam is admitted through suitable valves, to displace the oil at the speed required, and deliver it past sight-holes to the parts to be lubricated.

Fig. 30 shows a representative mechanical feed lubricator, which is operated by a rod receiving motion from one of the moving parts of the engine. The oil in the reservoir is forced by small pumps along the pipes leading to the different points of lubrication. The throw of the pumps can be so regulated that they deliver any desired quantity of oil, and as the barrel receives a regular supply a constant stream of lubricant is assured. In many instances the mechanical lubricator is supplemented by a steam jet to act as an anti-carbonizer.

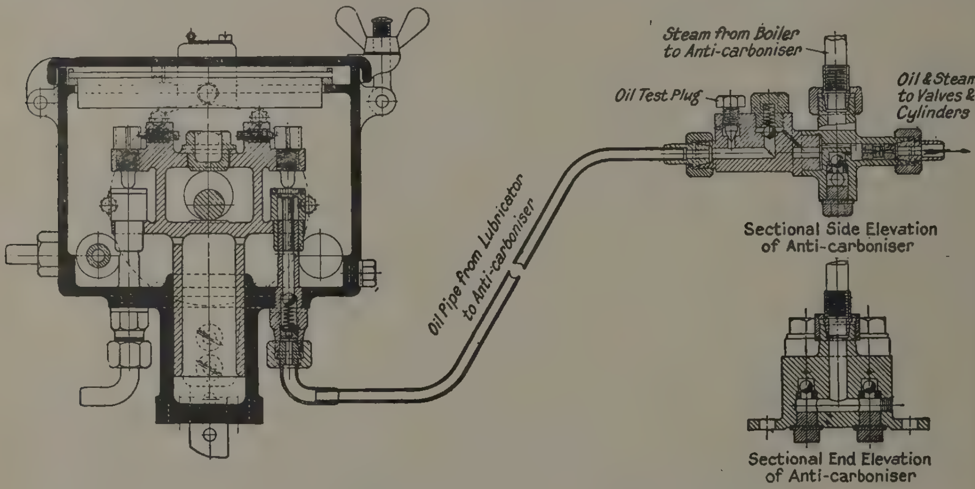
Steam from Boiler to Anti-carboniser
Oil Test Plug
Oil & Steam to Valves & Cylinders
Sectional Side Elevation of Anti-carboniser
Oil Pipe from Lubricator to Anti-carboniser
Sectional End Elevation of Anti-carboniser

Fig. 30. Wakefield Mechanical Lubricator and Anti-Carbonizer

Whilst the mechanical system of forced lubrication doubtless has its advantages in securing a known and definite delivery of oil, similar claims are made for the hydrostatic, or displacement, method, and Fig. 31 gives a typical form of such a device. In this, the oil chamber under pressure has a connection whereby condensed steam is admitted to it, the level of the water gradually rising in it displaces the oil, and forces this up through the different sight feed tubes connected to the pipes which convey it to the cylinders, etc.

There are many interesting varieties of these displacement lubricators, but it is quite impossible here to make mention of more than one example.

To guarantee satisfactory lubrication of the piston valves of superheater engines, arrangements are in use whereby on the opening of the main regulator to steam, displacement of the oil commences in the sight feed lubricator, ensuring automatic action.

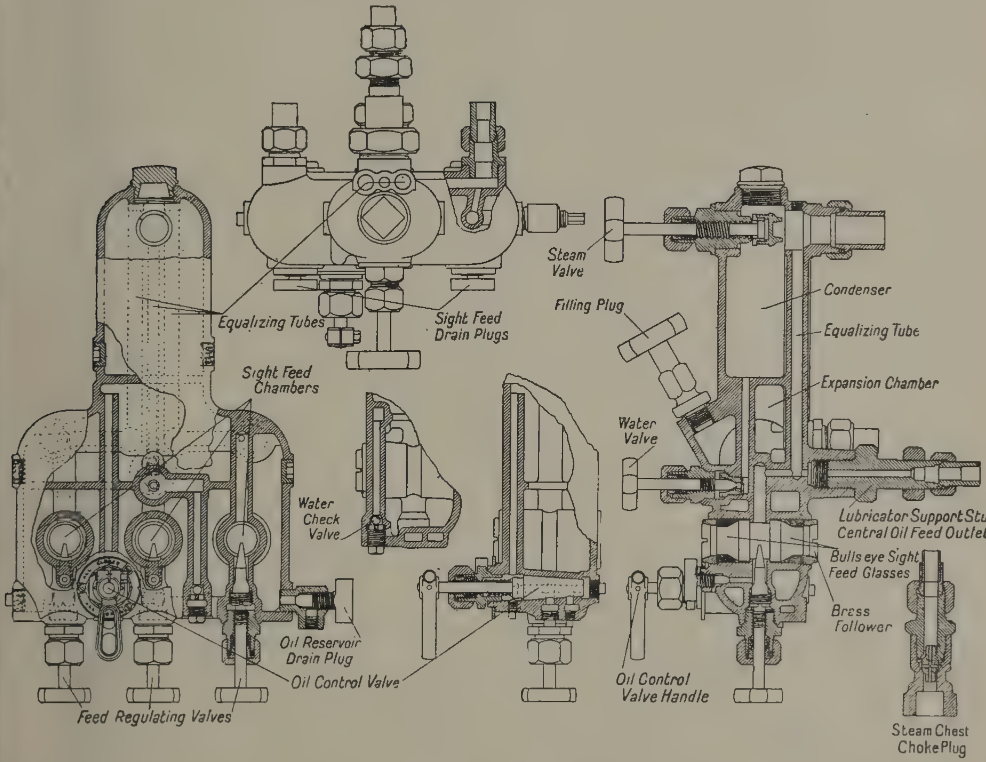

FIG. 31. DISPLACEMENT LUBRICATOR

The axle-box guides are details which require somewhat careful treatment in the way of lubrication, as much trouble may result from axle-boxes seizing in the guides if lubrication ceases. Special oil boxes are often provided at the side of the engine with pipes leading to the surfaces of these guides.

The axle-boxes are furnished with pads below, which have trimmings hanging down, or are partially immersed in the oil in the "keep," or oil well.

The moving parts of the motion, such as the coupling rod ends, big ends, slide bars, links, etc., are provided with oil cups in which worsted trimmings, or other devices are inserted to secure a gradual and continuous feed of oil by capillary attraction. To stop the action of such lubricators, it is only necessary to lift the trimmings from the pipes, which is usually done when the engine reaches the shed for standing off.

So far, oil of approved qualities has held the field in this country for the lubrication of locomotives; during recent years it has ousted grease from even the axle-boxes of wagons. Strange to say, something of a reverse nature has happened in America and some other countries. There grease has replaced oil, for the lubrication of the driving journals, big ends, and heavy moving parts of locomotives;

further, it has been adopted with marked success in certain tropical lands for the lubrication of the journals of rolling stock.

The application to locomotives has been carefully studied, and the preparation of a suitable grease, with the best and most satisfactory means of applying it, are the result of much investigation and experience. It would certainly appear that grease should be a more economical form of lubricant, for it is only when actual work is being performed that the medium will be used; there is little or no waste from leakage, drip, etc., and, further, it is much cleaner both in storage, application, and use. It is admitted, however, that the friction incurred is somewhat greater with grease than with oil, but it is not considered a sufficient detriment, especially when slow moving freight locomotives are concerned.

Graphite, used dry or mixed with oil, is also employed as a lubricant for locomotive valves and cylinders; it has also been tried for journals.

The consumption of oil in the lubrication of a modern locomotive naturally varies very considerably with the type of engine, service performed, etc. It is stated that on main line working on a British railway, a consumption is recorded of from 3·5 to 5 pints of engine oil, and from 2·5 to 3·5 pints of cylinder oil per hundred engine miles.

Superheater engines are said to require less cylinder oil than those using saturated steam, a fact to be attributed, probably, to the superior grade of oil used and its very high flash point.

In the actual delivery of the lubricant to the valves of cylinders, there is some differences in practice; some designers are content to partially disintegrate the oil on its passage into the steam pipe, whilst others insist on its delivery to the actual surfaces. Evidence can be adduced for the efficiency of either, and it probably depends more on the quality of the oil than in its particular delivery, for if the oil is carbonized, or only partially so in its transit, it does not matter much how the delivery is assured.

THE TESTING OF PRIME MOVERS

By R. Royds, M.Sc., M.I.Mech.E.

LESSON VII

BOILER TESTING

The remainder of our discussion will be concerned principally with the testing of boilers, steam-engines and turbines, and internal combustion engines. No very useful purpose would be served here by going into a great number of details, and only essential factors will be dealt with. For those more vitally interested who may require more detailed explanations, and who would like to have suitable forms for tabulating test results, reference may be made to the 1926 Report of the Committee on tabulating the results of *Heat Engine Trials*, published under the authority of the Council of Civil Engineers, by William Clowes & Sons, Ltd., and to the work on *The Testing of Motive Power Engines*, published by Messrs. Longmans, Green & Co.

Boiler Tests. A boiler test made for industrial purposes is usually concerned with the efficiency and performance of the boiler under definite conditions. The manner of making the test, the methods of measurement to be adopted, and the readings to be observed, depend upon whether one or more isolated tests are required, or whether the performance under the ordinary conditions of working over, say, a period of a week or more is to be ascertained. In the latter case the usual purpose is to know what the boilers are doing with a view of making proposals for improvements if the results call for improvement. In the former case the tests are usually concerned with the fulfilment of guarantees, or with obtaining an authentic report of the capabilities of the boiler plant with respect to efficiency.

To determine the efficiency of a boiler it is necessary to measure the fuel consumption, the calorific value of the fuel, the amount of steam generated, together with water and steam temperatures, steam pressures, condition of steam leaving the boiler, etc. If the test were of great importance it would be found advantageous to ascertain the measurable losses of heat due to lost fuel in the ashes, incomplete combustion, and the heat carried away by the flue gases. In this case the fuel and ashes would require to be analysed as well as the flue gases, and the temperature of these gases leaving the boiler plant, and perhaps at intermediate points as well, would have to be measured along with the atmospheric temperature. By demonstrating that the whole of the measured and calculated quantities of heat in the heat balance sheet are reconcilable, and that the heat given to the water plus the estimated losses do not exceed the estimated heat given by the fuel, the authenticity of the boiler test is greatly enhanced.

Duration of Tests. The duration of a boiler test should depend primarily upon the accuracy with which the finishing conditions can be adjusted to those obtaining at the start, more especially with respect to the thickness and condition of the fires when the furnaces are coal fired. If the coal is mechanically fired, as with a chain grate stoker, it is comparatively easy to arrange that the grate conditions at the finish are approximately the same as those at the start, provided the rate of working of the boiler is kept something like constant. When the coal is hand fired, however, the person in charge of the test has to judge the thickness and condition of the fire at the start, and has to try to stop the test with the same thickness and conditions. It is difficult to judge how much coal and how much ash and clinker there is on the grate and, therefore, the experimenter should try to make the duration such as to cover one or more cycles from clean fires to clean fires. If the fires are cleaned, say, half an hour before the test commences, and the last cleaning is about half an hour previous to the expected end of the test, the liability to error is much reduced. In no case should the duration of a boiler test be less than four hours. If the experimenter will have all the readings or observations plotted on a time base as the test proceeds, then the course of the plotted values will indicate whether something like constant conditions were being obtained during the test, and the official beginning and end of the test could be selected according to the conditions shown by the plotted observations.

Coal Weighing. The stack of solid fuel to be used during a test should be kept well away from the firing floor, and the fuel to be used would be weighed out periodically. A box, pan, or barrow on a weighing machine could be used for this purpose, weighing out, say, 200 lb. at a time. Only one weighing should be allowed on the firing floor, and the time this weighing is finished would be recorded so that the total weight of fuel fired up to any time during the test is readily ascertained. Some care is necessary in recording the weighings as otherwise one or more of the weighings might go unrecorded.

Sampling of Coal. An average sample of the fuel would be necessary for the determination of the calorific value and, if required, for purposes of analysis. The fuel, usually small coal, can be sampled by taking, say, one-quarter to one-half of a shovelful at each weighing and placing it in an air-tight canister until the end of the test. Then the whole of the samples thus collected would be emptied on to a solid floor or iron plate and rapidly mixed, and then formed into a square or pyramid. The bulk would be quartered and opposite quarters then rejected. The remainder would be similarly mixed, quartered, and rejected until a final sample is left of 2 or 3 lb. in weight, and this would be placed immediately into an air-tight bottle so that the moisture in the fuel will not be evaporated. The determination of the moisture contents, the calorific value, and an analysis of the fuel requires special skill, and should preferably be done by a qualified chemist as this test is important.

Measurement of Feed Water. In connection with the measurement of the feed water, the water level in the boiler drum, as shown by the gauge glasses, should be recorded periodically during the test, and, if possible, the finishing level should be arranged to be the same as at the start or the difference allowed for. The feed water is usually measured on the suction side of the feed pump, and observations should be made for possible leakages of water at the feed pump, feed pipe, boiler blow-off valve, and boiler seams, for if leakages occurred during the test, the boiler would be credited as having evaporated such water unless the leakages were allowed for.

The best method of measuring the feed water is to weigh it in a tank on a weighing machine before running it into the suction tank. An alternative method which is easier to manipulate is to provide two calibrated measuring tanks placed over the suction tank as shown in Fig. 38. The upper tank has a water-tight division wall *A*, and each compartment thus formed is provided with some form of overflow pipe *B* and emptying cock *C*. The feed water is supplied alternately into the two upper measuring tanks so that one is filling while the other is emptying into the suction tank *S*. Some form of gauge, such as the hook gauge *H*, would be used to show a standard level in the suction tank *S*. To avoid mistakes it is important to record each filling as soon as it is emptied into the suction tank, but the time recorded against each filling would be that at which the level in the suction tank next falls to the standard level. By this method the amount of feed water measured up to any time during the test is readily ascertained. The amount of water required to fill each measuring tank could be obtained either by use of a standard gallon measure or by emptying into them weighed quantities of water.

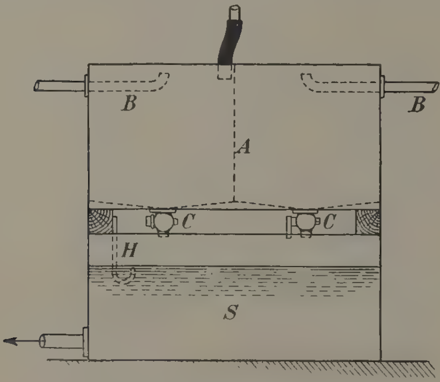

FIG. 38. WATER MEASURING TANKS

In power stations it is the usual practice to have continuous records of the fuel and feed water consumptions under running conditions. The fuel, say coal, is measured by a meter, such as the Lea coal meter. The water may be recorded by means of a Lea recorder or by a Kent Venturi meter. If the feed water is taken from a main having an ordinary water meter in circuit, and only this water is flowing through the meter during a test, then for approximate or comparative results the water may be measured in this way provided the meter has been calibrated recently. Again, a steam meter of reputable make placed in the steam pipe from the boiler could be used to give the approximate amount of steam produced.

In a boiler test the draught at one or more places in the boiler or flues is usually measured, especially if forced or induced draught is used. It may be that one of the objects of the test or tests is to ascertain or to demonstrate the rate of heat transmission or rate of evaporation under different draught conditions. For the laws connecting the heat transmission and the draught, reference may be made to the books on *Heat Transmission by Radiation, Conduction, and Convection*, and *Heat Transmission in Boilers, Condensers, and Evaporators*, published by Messrs. Constable & Co.

The efficiency of a boiler, including or excluding any superheater, economizer, or air heater as the case may be, is the ratio between the amount of heat in the steam generated, reckoned from feed temperature, per pound of fuel fired, and the calorific value of the fuel. If the boiler delivers saturated steam the dryness fraction of the steam may be ascertained by means of a steam throttling calorimeter. Not only is it important, therefore, to measure the quantity of the feed water, fuel, and temperatures, etc., as accurately as possible, but it is equally important to obtain a true average sample of the fuel and to determine the moisture contents and calorific value of the fuel with a reasonable degree of accuracy.

The character of the combustion and the amount of air supplied to the furnace per pound of fuel burned are usually estimated from the analyses of the flue gases and the fuel.

WORKS ORGANIZATION AND MANAGEMENT

By E. W. Workman, B.Sc., A.M.I.E.E., etc.

LESSON XII

THE TIME-KEEPING DEPARTMENT

Scope of Work. As was mentioned in the last lesson, the work of the time-keeping or wages department is very often done by one section of the cost department. Whatever the organization in this respect, however, the work done is substantially the same. The objects of the work are two—

1. To ascertain the correct wages payable to each employee.

2. To analyse these wages so that it is known to the cost department how much should be charged to each order and to the different overhead expense accounts.

The statistical summaries of the wages paid out may also be prepared by the wages section of the cost department, but the actual making up and paying out of the wages to the employees should be done by the cashier's department. It is important that wages should be calculated and paid by entirely different departments in order to minimize the possibility of fraud and theft.

Time Records. There are two main requirements that have to be met in the records of working hours which are compiled by or for each employee.

1. A record must be kept of the *total* time worked by each employee each week in order to enable day wages to be calculated, and so that a check may be kept on absentees and late comers.

2. Details must be kept of the work on which each employee has been engaged, partly in order to calculate piece-work wages, and partly so that the correct amount of labour cost may be charged to each individual job or process.

Pay-roll Clock Cards. The total time worked by each employee each week is sometimes compiled from records kept at the entrance gate of the time when each employee receives a metal disc on entering the works, and of the time when the disc is returned when the employee leaves the works later on. More usually these records are obtained by means of automatic recording clocks which print the times of entering and leaving on cards bearing the employees' names and numbers. At the end of each week the clock cards are added up to show the total number of hours spent in the works by each employee, and are used in conjunction with the job tickets described below to calculate the wages due for the week. Although the printing of the time is quite automatic when a card is inserted in a clock, the employees should not be left to record their time entirely without supervision, as otherwise there is nothing to prevent the marking by friends of the cards of absentees or late comers as though they had worked the full time and then later drawing the full wages—though this does not, of course, apply to piece-workers. The names and numbers of the employees are conveniently put on the cards in advance by means of an addressing machine used also for the pay-roll and pay envelopes.

Job Tickets. The details of the work performed by each employee are recorded on some form of job ticket, the design of the form depending on the particular conditions of the work. In some cases it is convenient for the workman to state on one job ticket the length of time taken for each job worked upon during the week, but more often a separate job ticket is prepared for each job worked upon by each employee each week. In this case, at the end of each job the employee fills up a printed job ticket giving his name and number and the particulars of the job number, the description of the work and the amount of time taken. In the case of work being done for definite order numbers for customers, the job or order number is quoted on the ticket, but with indirect workers, such as sweepers-up or store-keepers, a special "expense number" is quoted, the wages for these workers being charged to overhead expenses instead of to individual jobs. In the case of piece-work, the ticket should be passed and signed by an inspector for certification as to the quantity and quality of the work done.

Comparison of Clock Cards and Job Tickets. Both the clock cards and the job tickets should each give a complete account of the time of each employee, and so the total of the times shown on the individual job tickets should add up to the time shown in total on the clock card. Such a comparison should always be made as it forms a useful check that no day work job tickets are missing. As day wages will usually be computed from the clock cards there is always a tendency to avoid making out the corresponding job tickets, and if this is not checked the jobs affected will be undercosted by the value of the hours not recorded.

Pay-rolls. The number of hours of day-work recorded on the clock cards or job tickets are multiplied by the day-work rate of the employee concerned, in order to give the basis amount of day wages payable. Similarly, the number of completed pieces are multiplied by the respective piece-work prices in order to give the basis amount of piece-work wages payable. To these basis amounts are added or subtracted any overall alterations to the rates of wages which are required by national or other agreements, and the total result for each employee is then recorded on the pay-roll. This form merely consists of a convenient summary of the wages payable. It is divided up into a number of vertical columns giving, respectively, the employee's clock number and name, his gross earnings, and any deductions for insurance, club, etc., together with the net amount payable. The different columns are added up and a " cash analysis " is made of the total, to show the quantity of each coin required to make up the individual wages. The sheets should be fully signed by every clerk concerned in their compilation, and also by the head of the department. After this they should go for scrutiny and signature by the foremen and superintendent of the department to which the pay-roll refers, after which they are ready for the cashier.

Paying Out Wages. On receipt of the pay-rolls, the cashier's staff count out the cash payable to each employee, and place it either in an envelope or tin receptacle bearing the clock number of the employee. In the meantime the wages due to each employee with details of any deductions have been entered by the wages department at the top of each clock card, and the clock cards returned to their owners. This enables each employee to check that he is receiving the correct amount of wages. The presentation of the clock card also forms a convenient form of identification to the paying-out clerk, who obviously cannot be expected to know all the men's numbers. Payment may be made on presentation of the clock card at the cashier's office, but is more usually done in the shops by the cashier visiting each department in turn with the corresponding pay packets. Both the cashier and a witness should sign the pay-roll, and any packets left over should be returned to the cashier's office until claimed.

The advantages of using transparent pay envelopes are that the money can be checked by both the cashier and the employee without opening the packet, and that the money can be more easily handled when made up in that form. Where transparent pay envelopes are used a rule should be made that any mistakes in the cash will not be rectified unless the pay envelope is returned unopened. As pointed out above it is important that different people should make up the pay-roll, count the money into the envelopes and actually pay out the wages.

Filing. It is of the greatest importance that sound filing methods should be adopted by the time-keeping office and by the cost department generally. Back records are invariably wanted when a dispute arises or even for ordinary investigation, and more time will often be wasted in trying to find badly filed documents than will suffice to put the whole lot in proper order. In some works filing is of extraordinary importance as detailed costs are not worked out in every case, but all requisitions and job tickets are filed under their respective order numbers until there is some reason for getting out the cost of a particular job in detail. Such a method will prove a great saving in clerical expenses if it can be adopted, but the results will be fatally incomplete and misleading if the filing is anything else than perfect. At the same time, the accumulation of old papers is liable to become so great in a large factory that it is absolutely imperative to lay down some scheme whereby papers of different ages in different classifications may be automatically destroyed, unless especially marked for preservation. The classified summaries of the results may be kept, of course, for a much longer period, long after the date when it is possible to destroy the details from which they have been compiled.

PUMPS: CONSTRUCTION AND MAINTENANCE

By Owen A. Price, M.I.Mech.E.

LESSON IX

THEORY OF THE CENTRIFUGAL PUMP

Proceeding with our endeavour to obtain physical conceptions of the behaviour of moving fluid we shall study bodies of water in rotation. Commencing with a simple confined *forced vortex* we shall see how other types of vortex may be produced and what are their properties. From these considerations the behaviour of a centrifugal pump will emerge naturally, and a tangible interpretation will be found for each term in the theory.

Simple Forced Vortex. Imagine a closed cylindrical chamber (Fig. 33) of radius r_2 ft. filled with water which is stirred uniformly by a paddle or impeller. Then, in effect, the water rotates as a solid mass, all particles revolve in circular paths and each has the same angular velocity, say, ω radians per second. Such a body of rotating fluid forms a simple *forced vortex*, with the whirling velocity of every particle directly proportional to its distance from the axis. This whirling velocity w ft. per second at any radius r ft. can, therefore, be written as $w = \omega r$ ft. per second, which, by expressing the relation between the whirling velocity and the radius, enables us to calculate the *rate of change* of pressure along a radial line. Thus, using the expression derived for "Curved Streams"

$$\frac{dh}{dr} = \frac{w^2}{gr} = \frac{\omega^2 r^2}{gr} = \frac{\omega^2 r}{g}$$

from which the pressure, or centrifugal head h ft., at any radius is obtainable by integration of the pressure to that radius. Thus, integrating from zero radius to radius r_2,

$$h_2 = \frac{\omega^2 {r_2}^2}{2g} = \frac{{w_2}^2}{2g} \text{ ft. head}$$

which shows that *the lateral pressure, or centrifugal pressure head, at the circumference of a forced vortex through which no water is flowing, is equal to the head due to the circumferential velocity.*

The building up of pressure within the vortex in a radially outward direction is illustrated by the profile assumed by the free surface (see Fig. 33) when such a vortex is formed in an open vessel. Each particle not only exerts its own centrifugal force but has transmitted to it the centrifugal force, in the form of centrifugal pressure, of all the inner particles along the same radial line, the radial banking up of the rotating surface being an indication of the cumulative increase of centrifugal pressure, and the parabolic profile representing an increase of pressure head proportional to the square of the whirling velocity.

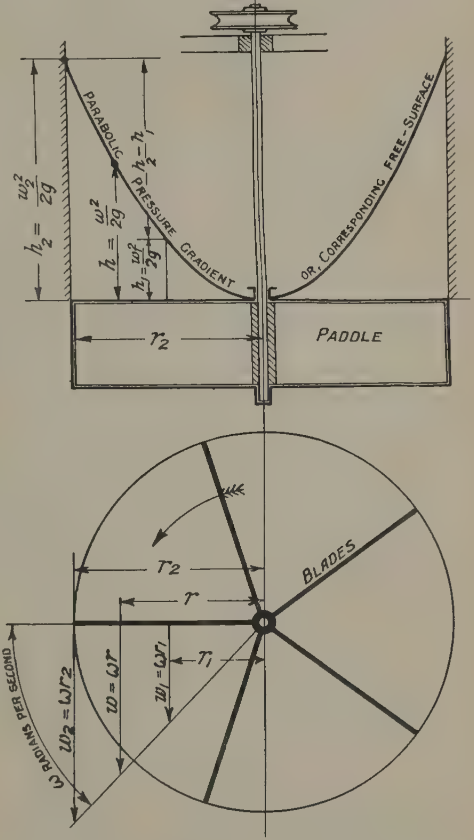

Fig. 33. Confined Forced Vortex

Obviously, also, the difference in pressure $(h_2 - h_1)$ ft. head between any two radii, r_2 ft. and r_1 ft., within the forced vortex is obtainable by integration of the pressure between those two radii, thus

$$h_2 - h_1 = \frac{\omega^2}{g}\left(\frac{r_2^2 - r_1^2}{2}\right) = \frac{w_2^2 - w_1^2}{2g} \text{ ft. head.}$$

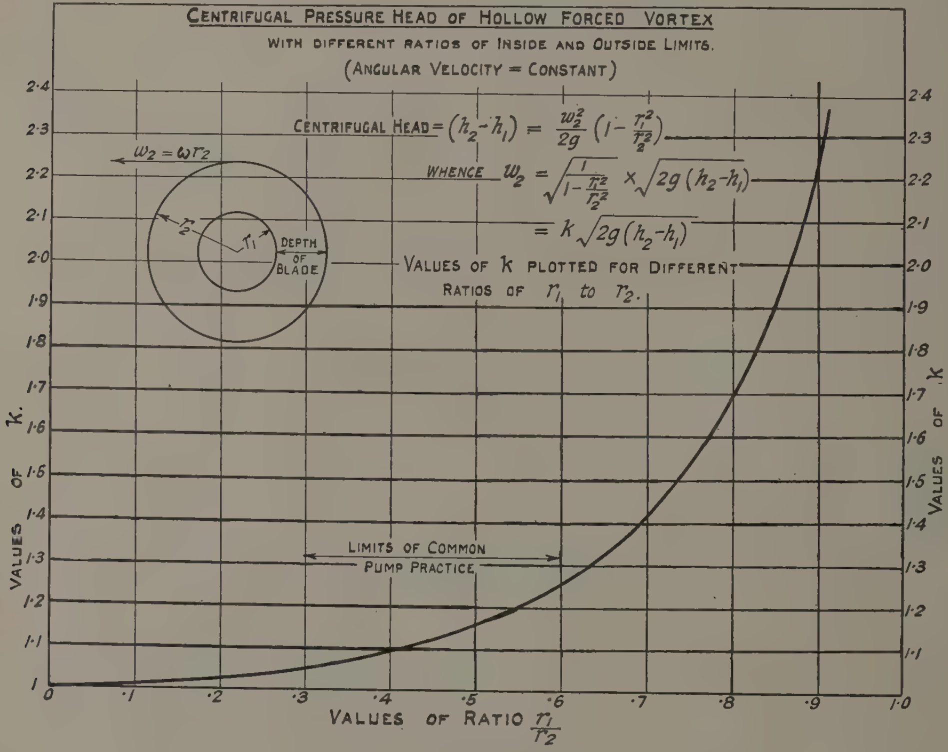

FIG. 34. HOLLOW VORTEX COEFFICIENT

Hollow Forced Vortex. With the majority of centrifugal pump impellers (see Fig. 27) the blades do not extend inwards to the axis, and such blades form a *hollow forced vortex* in which no rotational energy is impressed on the water forming the core. The resulting centrifugal pressure is given by the above expression with r_1 as the inside radius and r_2 the outside radius of the blades. Therefore, we find that, *if the forced vortex is hollow the centrifugal pressure head at the outer circumference decreases as the size of the core increases, and is equal to the difference between the pressure of a full vortex and the pressure of a smaller vortex equal in size to the core and rotating at the same angular velocity.*

In the expression just given, w_1 and w_2 may be related by introducing the ratio $\frac{r_1}{r_2}$ and writing the equation for centrifugal head in the form

$$h_2 - h_1 = \frac{w_2^2}{2g}\left(1 - \frac{r_1^2}{r_2^2}\right) \text{ ft. head}$$

whence $w_2 = k\sqrt{2g\,(h_2 - h_1)}$ ft. per second

where $k = \sqrt{\dfrac{1}{1 - \dfrac{r_1^2}{r_2^2}}}$

The curve of Fig. 34 shows values of k for a range of values of the ratio $\frac{r_1}{r_2}$, and demonstrates that k may have any value from unity for a solid vortex with $\frac{r_1}{r_2} = 0$, to infinity when $\frac{r_1}{r_2}$ approaches 1, and the depth of the blade becomes exceedingly small. Therefore the whirling velocity for any given centrifugal head may have any value from $\sqrt{2g\,(h_2 - h_1)}$ feet per second to infinity according to the value of $\frac{r_1}{r_2}$.

A pump impeller creates an approximation only to a hollow vortex because the central core is not stagnant water, but is rotated at an appreciable velocity by the revolving pump spindle, impeller boss, impeller shrouds, and inner vane tips.

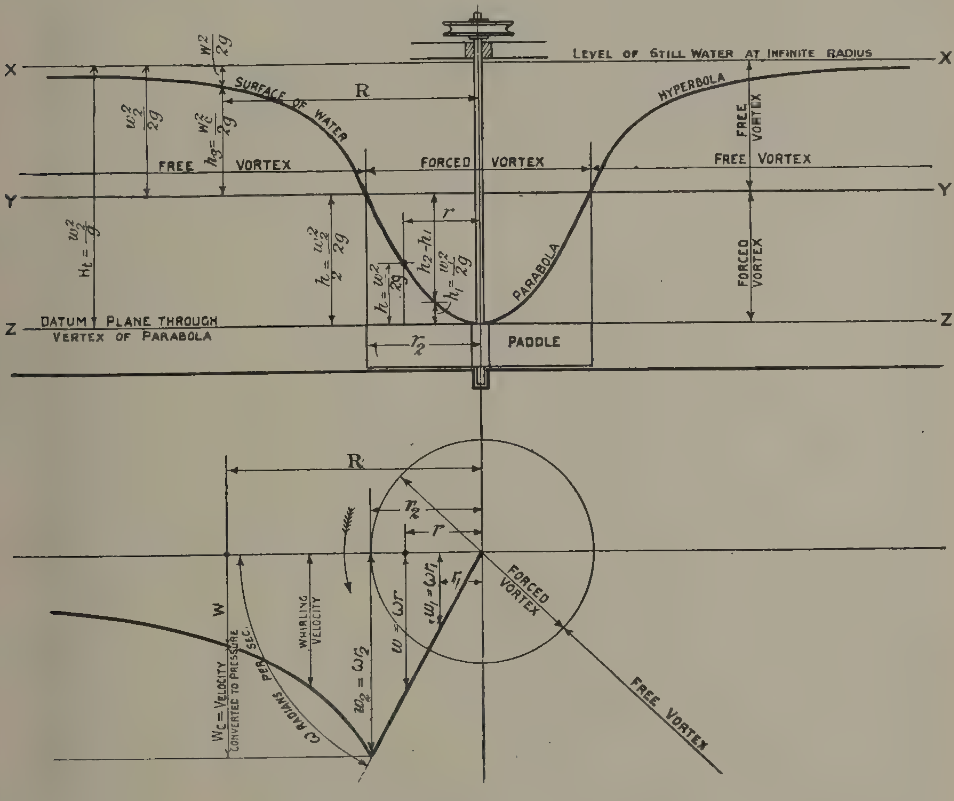

FIG. 35. TOTAL ENERGY IN A FORCED VORTEX

A true hollow vortex, however, can be formed by admitting air to the core, though the result is then of no value for pumping.

Total Energy Head in a Forced Vortex. The centrifugal pressure in a vortex is the *lateral pressure* arising from the curvature of the lines of motion, as explained in Lesson VIII. Hence we may say that *the centrifugal pressure in a vortex is the static head which the rotating fluid can support in a direction at right angles to its own motion*, as is evident from our expressions and as shown in Fig. 33. In addition to this lateral pressure the fluid possesses angular momentum in consequence of its velocity of whirl.

Consider a forced vortex surrounded by a large volume of free water so that the whole of the energy of motion is absorbed and reduced ultimately to static energy. Thus in Fig. 35

the forced vortex (similar in every respect to that of Fig. 33) is contained within the radius r_2, and the whole body of fluid is supposed to be included in an indefinitely large open circular vessel.

At the limiting radius r_2 of the forced vortex the centrifugal pressure as before is $h_2 = \frac{w_2^2}{2g}$ ft. head, and the water has a velocity w_2 ft. per second. No energy is transmitted to the free water beyond radius r_2, and this outer water therefore is a *free vortex* with an induced rotation by virtue of the angular momentum impressed at the cylindrical surface of juncture with the forced vortex. Disregarding friction losses, no energy can be lost, and therefore the impressed angular momentum (or moment of momentum) about the central axis must remain unaltered throughout the free vortex. Since the moment of momentum of unit mass M of water at radius r_2 is Mw_2r_2, therefore

$$Mw_2r_2 = M.W.R. = \text{constant}$$

$$\text{and} \quad W = \frac{w_2r_2}{R}$$

where W ft. per second = the circumferential velocity of whirling and R ft. = the radius for *any* point in the free vortex. Whence it is evident that W varies as $\frac{1}{R}$ or, *the velocity of whirling in the free vortex is inversely proportional to the distance from the axis.* Also, as energy is neither given out nor added to the free vortex, or, by reason of the freedom of the vortex, any inequality of distribution would be immediately equalized by the free intermixture of the particles in adjoining circular currents, hence *the distribution of energy is uniform throughout and the total energy per unit of mass is constant at all points.*

The free surface of the water is a surface of equal pressure (atmospheric), and any change of pressure over a horizontal plane passing through the vortex will be indicated by a change of level at the water surface.

To determine the lateral pressure changes h_3 through the free vortex over a horizontal plane, we can apply the principle of Bernouilli's theorem. That is, the total energy (H_t ft. head) is constant throughout and equal to the sum of the pressure, elevation, and kinetic heads, thus, relative to the datum plane ZZ passing through the axis and touching the surface of the water at its lowest point,

$$H_t = h_3 + h_2 + \frac{W^2}{2g} \text{ ft. head} = \text{constant}$$

or,

$$H_t = h_2 + \left(\frac{w_2^2}{2g} - \frac{W^2}{2g}\right) = h_2 + \frac{w_2^2}{2g}\left(1 - \frac{r_2^2}{R^2}\right)$$

ft. head, whence, if the free vortex extends indefinitely, R becomes infinitely great and, in the limit

$$H_t = h_2 + \frac{w_2^2}{2g} \text{ ft. head.}$$

Whence the total energy head created by a forced vortex is given by

$$H_t = 2\left(\frac{w_2^2}{2g}\right) = \frac{w_2^2}{g} \text{ ft. head,}$$

therefore, *the total energy of any particle in a solid forced vortex is half kinetic and half potential.*

As every particle in the rotating mass—both forced and free vortex—describes a circular path, there being no radial flow, the energy transmitted and the velocity distribution are not affected in any way by the inclination of the impeller blade to the radius. Applying the foregoing results to the centrifugal pump, it is clear that when there is no flow through the impeller the centrifugal head is determined by the diameter of the blades regardless of the curvature of the vane. Also, if the shape of the pump casing is unsuitable for the formation of a partial free vortex (as is usually the case), none of the kinetic head at the periphery of the impeller is recovered, and the pressure head given by the pump will be approximately,

$$h_2 = \frac{w_2^2}{2g} \text{ ft. head.}$$

In this expression we recognize Denis Papin's (year 1689) fundamental relation between head and peripheral velocity in a centrifugal pump, and to which reference was made early in Lesson VII.

Radial Flow through a Forced Vortex. Reverting to the cased-in impeller of Fig. 33, it is plain that if a hole were drilled in the casing a jet of water would escape whilst the paddle was in rotation. It would not be unreasonable to expect that as the jet possessed energy the pressure in the casing would be lowered by the escape of water, also that the larger the hole the greater would be the escape of water, and the greater the reduction of pressure inside the casing. Though this is generally true, it is not necessarily so as regards the pressure drop. Whilst the jet is discharging there is *radial flow through the forced vortex*, disturbing the concentric paths of motion of the particles, and altering them to outward growing spirals.

With *radial flow* through a forced vortex the velocity of every particle is compounded of two velocities, viz. the velocity along the blade, or vane, commonly called the *relative velocity* (i.e. "relative" to the vane), and the velocity of revolution, usually termed the *velocity of whirl*, the relation between these two components being determined by the inclination of the blade to the radius. Thus, referring to Fig. 36, the outer tip of the impeller vane may be radial (*A*) bent forward (*B*) relative to the radius, or bent backwards (*C*, *D*, and *E*) relative to the axis, and the resulting effect on the ratio between the centrifugal pressure and the velocity energy is different for each case. In these diagrams v_2 represents the relative velocity, u_2 is the vane tip velocity, and the resultant of the two is V_2, usually termed the *absolute velocity* of the water. For computing purposes it is necessary to resolve V_2 into a radial component of flow f_2, and a tangential velocity of whirl w_2. Inspection of the diagrams informs us that the velocity of the water is no longer identical with the velocity of the blade, and that *when radial flow occurs through a forced vortex the centrifugal pressure head is determined by the velocity of whirl of the water and the kinetic head by the absolute velocity of the water.* We are also able to discern three distinct varieties of forced vortex produced respectively by the three types of vane, radial, forward-bent, and backward-bent tip, in which, when radial flow is in progress, the angular velocity of the water is respectively equal to, greater than and less than the angular velocity of the driving vane. Further variations occur with different angularities of backward bent vane tips.

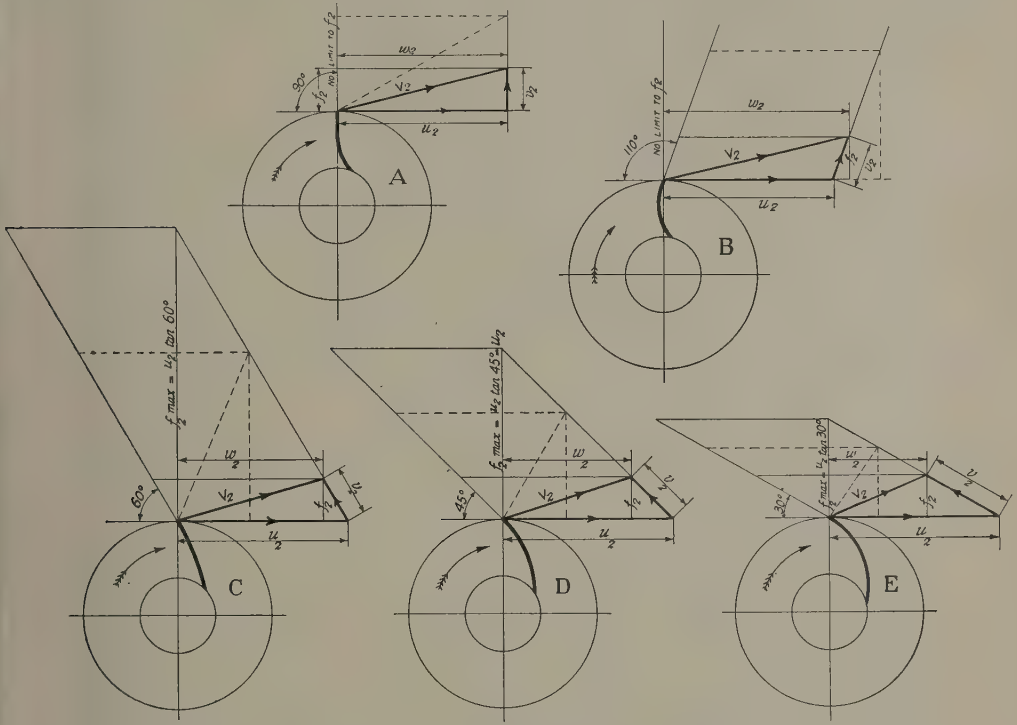

FIG. 36. FLOW THROUGH A VORTEX—VANE TIP VELOCITY DIAGRAMS

By calculating the whirling velocities and absolute velocities for different rates of flow the curves of Fig. 37 have been plotted showing the effect of radial flow on the centrifugal pressure head $\frac{w_2^2}{2g}$, the kinetic head $\frac{V_2^2}{2g}$, and the total energy head H_t for forced vortices with the five vane tips shown in Fig. 36. The impeller diameters and peripheral velocities are identical for the five cases, and the centrifugal pressure head and kinetic head have both been taken as 100 for the condition when no flow occurs; the radial velocity of flow f_2 is plotted as a percentage of the peripheral velocity of the vane tip throughout, and therefore, the several diagrams are strictly comparable.

The main features, evident from the diagrams, are tabulated below for each of the various types.

The shape and influence of the inlet edges of the vanes at the eye of the paddle, or impeller, can only be discussed with reference to a given rate of radial flow through the forced vortex and, therefore, this part of the subject must be deferred until a complete centrifugal pump is considered.

WITH NO RADIAL FLOW ($f_2 = 0$) THROUGH THE VORTEX

$w_2 = V_2 = u_2$ for all vane tip angles

$\therefore$ the centrifugal pressure head $\frac{w_2^2}{2g} = \frac{u_2^2}{2g}$

and the kinetic head $\frac{V_2^2}{2g} = \frac{u_2^2}{2g}$

WITH GRADUALLY INCREASING RADIAL FLOW THROUGH THE VORTEX

Figs. 36 and 37	Type of Vane	Limiting Flow f_2 through Impeller	Centrifugal Pressure Head $= \frac{w_2^2}{2g}$	Kinetic Head $= \frac{V_2^2}{2g}$
A	Radial tip	No limit	Remains constant at all flows Without limitation	Increases with increased flow No limit to increase
B	Forward bent tip	No limit	Increases with increased flow indefinitely	Increases with increased flow, but at a greater rate than with radial tip. No limit to increase
C	Backward bent tip α greater than 45°	$f_2 = u_2 \tan \alpha$	Decreased gradually to zero at maximum flow	Reduces to a minimum and then increases to a maximum as centrifugal pressure reduces to zero
D	Backward bent tip $\alpha = 45°$	$f_2 = u_2 \tan 45° = u_2$	Decreased gradually to zero at maximum flow	Maximum at no flow and decreases to a minimum, then increases again to same maximum
E	Backward bent tip α = less than 45°	$f_2 = u_2 \tan \alpha$	Decreased gradually to zero at maximum flow	Maximum at no flow decreases to a minimum and then rises slightly as centrifugal pressure reduces to zero

Where α is the inclination of the vane to the tangent at the impeller circumference.

ERRATA.

Lesson II, page 90, top.

The D'Arcy formula should read

$$I = \left[K\left(1 + \frac{1}{12d}\right) \times \frac{4l}{d}\right] \frac{v^2}{2g}.$$

Lesson VIII, page 958, bottom.

$\frac{\omega^2}{r}$ should read $\frac{w^2}{r}$

Lesson VIII, page 959, last column in lesson,

$\frac{\omega^2}{r}$ should read $\frac{w^2}{r}$ in both cases,

$\frac{\omega^2}{gr}$ should read $\frac{w^2}{gr}$.

"Whirling velocity ω" should read "whirling velocity w."

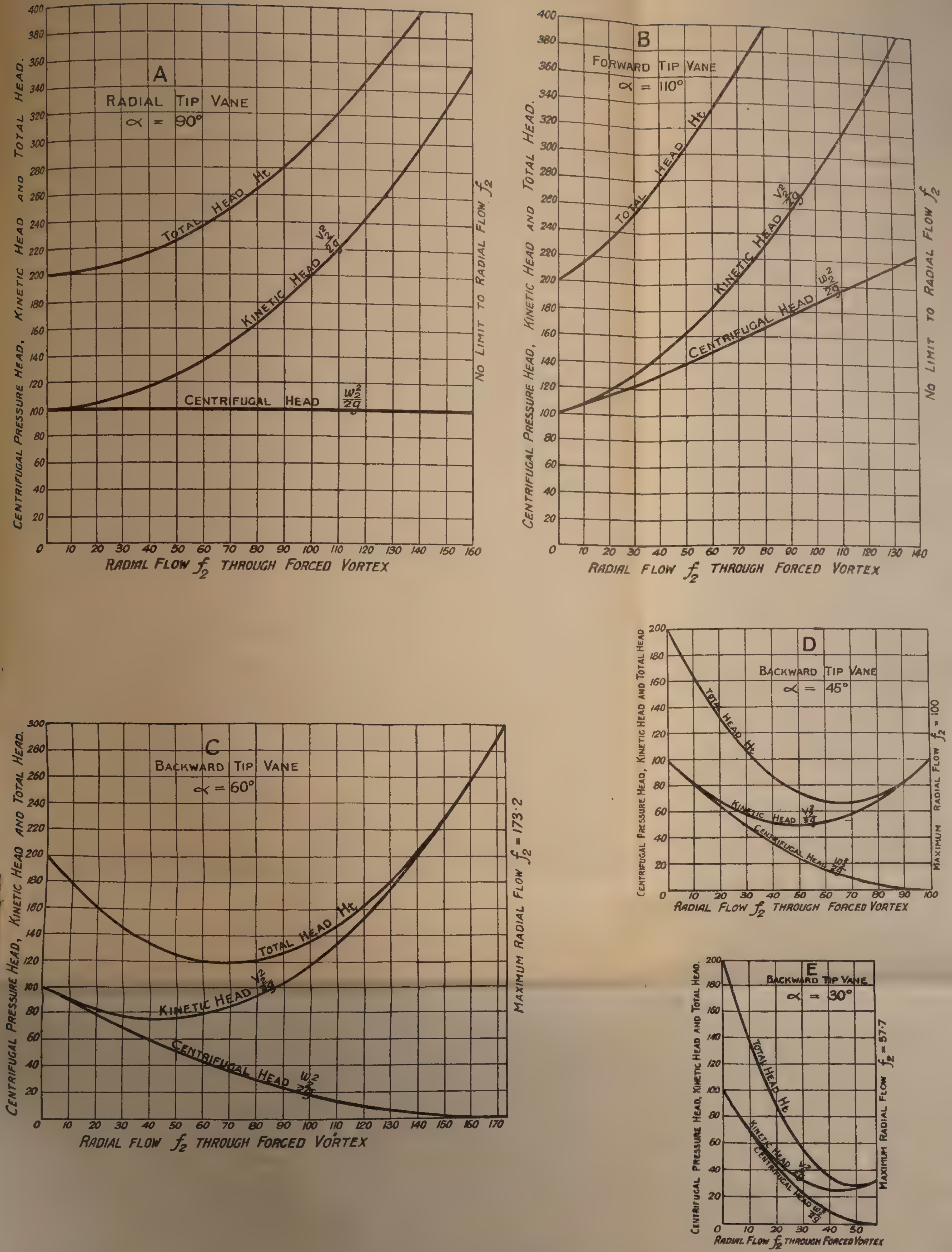

FIG. 37. ENERGY CHANGES WITH FLOW THROUGH A FORCED VORTEX

METALLURGY FOR ENGINEERS

By James Knowles

LESSON X

THE OPEN HEARTH—(*contd.*)[1]

Acid Practice

The acid method of open hearth steel making is so called because the hearth in which operations are conducted is composed largely of anhydrous silicic acid (SiO_2).

Before a hearth can be put in a new furnace the whole must be thoroughly dry. To this end fires are kept going in the regenerator chambers for a couple of days before the chequers are built, and smaller ones kindled near the manholes of all flues. The column of cold air in the chimney is removed by a pilot fire laid on a false grate at the base of the stack, and a coal fire is then built on the furnace bottom. Both gas valves are shut down meanwhile, and the reversing valve put on the centre so that the products of combustion pass through all the regenerators, the necessary air being drawn through the open charging doors.

Before admitting gas, precautions must be taken to prevent the formation of an explosive mixture in the regenerators or an accumulation of unburnt gas in the furnace or exit chequers. To ensure ignition the gas is tested for inflammability, the end on which it is decided to work thoroughly heated, and blazing wood provided near the entry ports.

When gas is first admitted no air is drawn through the valve, but is supplied through the doors. These are lowered as the gas becomes hotter, and the air valve gradually raised until normal working conditions obtain. In order to attain a good temperature in the exit chequers several hours elapse before the first reversal is made, but afterwards the intervals are progressively lessened until regular periods of 20 minutes to half an hour are adopted.

The Charge. Most specifications for acid Siemens steel call for low sulphur and phosphorus contents, a common maximum allowance being 0·04 per cent, and it is unusual for this to be extended beyond 0·06 per cent even for dead soft qualities.

The composition of the charge depends to a large extent on the specification, but other almost equally important factors are the cost of materials and the abundance, or scarcity, of suitable scrap. Whilst the pig-iron to scrap, ratio may be anything from 75 : 25 to 25 : 75 a proportion of 60 : 40 may be considered a good average. One hundred per cent pig charges are not unknown, however, and during the war many "all scrap" heats were worked along with some carbonizing material, such as anthracite. The excess pig in the charge is largely controlled by the sulphur content and, generally speaking, the greater the proportion of pig the longer will be the melt, owing to the larger percentage of carbon present.

Sulphur increases during melting due to bath concentration and absorption from the producer gas, whilst phosphorus is not reduced. If the notes in Lesson VIII, regarding the influence of temperature on the relative affinity of silicon and carbon for oxygen, are remembered, the importance of limiting the initial silicon content to about 0·75 to 1·0 per cent will be appreciated.

The pig-iron generally used is a mixture of Bessemer hematites Nos. 1, 2, and 3. Now, with English hematites, it is almost impossible to obtain regular and plentiful supplies of low silicon, low sulphur grades; when the latter is low the former is high and vice versa. Hence, if S and P contents of less than 0·04 per cent are required, a fair proportion of Swedish white pig may be included in the charge. The scrap introduced should be as near as possible to the chemical composition of the desired steel and suitable as to mechanical condition, very light scrap tends to increase the sulphur.

To assist in giving a clear idea of the compounding of charges, the following typical analyses of various constituents are given—

	West Coast Hematite	East Coast Hematite	Swedish White	Scrap
	%	%	%	%
Combined Carbon	·50	·45	3·75	·20
Graphite . .	3·50	3·50	—	—
Silicon . . .	2·40	2·25	·16	·04
Manganese .	·30	·75	·30	·55
Sulphur . .	·02	·03	·01	·05
Phosphorus .	·03	·05	·02	·06

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[1] The author is indebted to The Park Gate Iron & Steel Co., Ltd., for the loan of and permission to reproduce the photographs illustrating this lesson.

A 50 : 50 charge consisting of the East Coast iron and scrap mentioned above would run: C 2·07, Si 1·15, Mn 0·65, S 0·04, P 0·055. This may, perhaps, be considered a little high in silicon and could be rectified by using a larger proportion of scrap, but an increase in sulphur and phosphorus would automatically follow. Conversely, a lower sulphur and phosphorus could be obtained by using more pig.

Charging. The tap hole is made up by inserting a plug of anthracite from the outside, which is held in position by a stopper until the inside is filled with small coal. This is covered with dry white sand, and the outside filled up with anthracite breeze and undried white sand and finally rammed with loam. The hearth must be soaked with heat, and not merely superficially hot, before charging is commenced, otherwise the difficulties associated with cold working will be encountered. The initial temperature should at least be in the neighbourhood of 1,550° C.

There are several methods of charging, of which the following may be mentioned—

1. British: all the pig and then all the scrap. The scrap, being quickly oxidized is charged last to obviate the danger of it eating into the hearth.

2. American: the reverse of the foregoing. In this case quick melting is essential.

3. A mean of both—half the pig, half the scrap, remainder of pig, remainder of scrap.

4. Pig charged at intervals, with accompanying ore additions after the first charge is melted, the scrap being gradually added to the molten pig.

In the larger furnaces charging is done at intervals to avoid a serious reduction of temperature, and may be carried out by hand or mechanically.

Hand charging, slow and laborious though

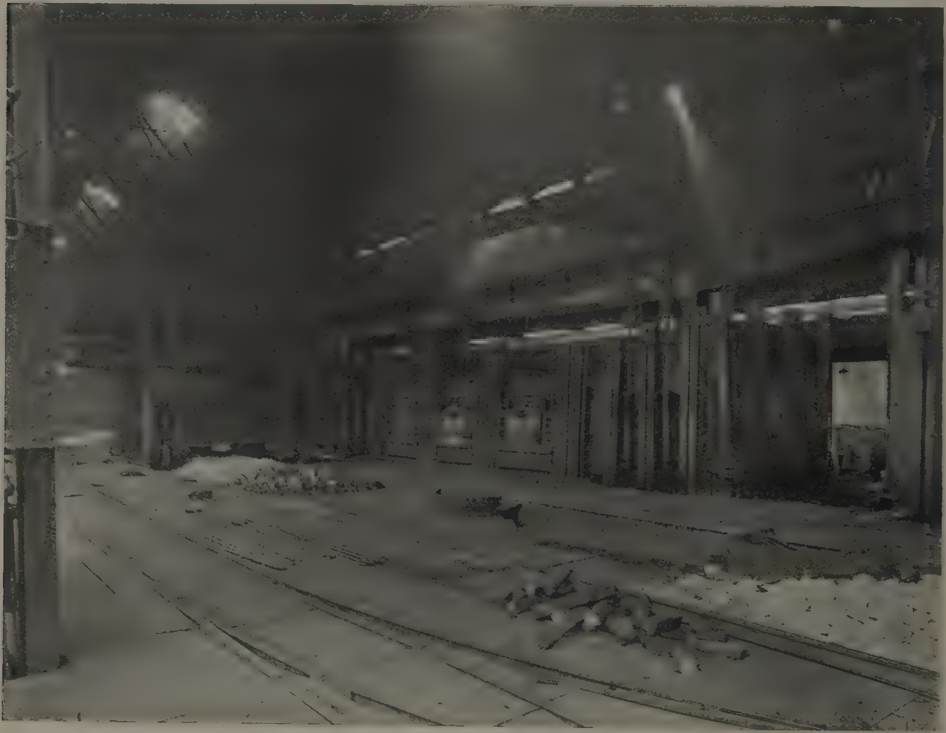

FIG. 15. CHARGING SIDE OF SIEMENS FURNACE

it may be, is the method common to most furnaces up to 30 tons. This involves the use of a tool known as a " peel," which is a long steel bar flattened out at one end so as to carry a half pig, with a loop handle at the other. The furnaceman supports the flat end on the jamb of the door whilst the piece is put on, then rapidly slides the peel into the furnace and projects the half pig into the desired position. A charge, neatly arranged so that all portions of the banks and hearth are protected by iron against the cutting action of the scrap, must be seen for the surprising skill of the workmen to be fully appreciated. Heavy scrap is similarly charged, whilst lighter varieties and the ore are thrown in with a shovel.

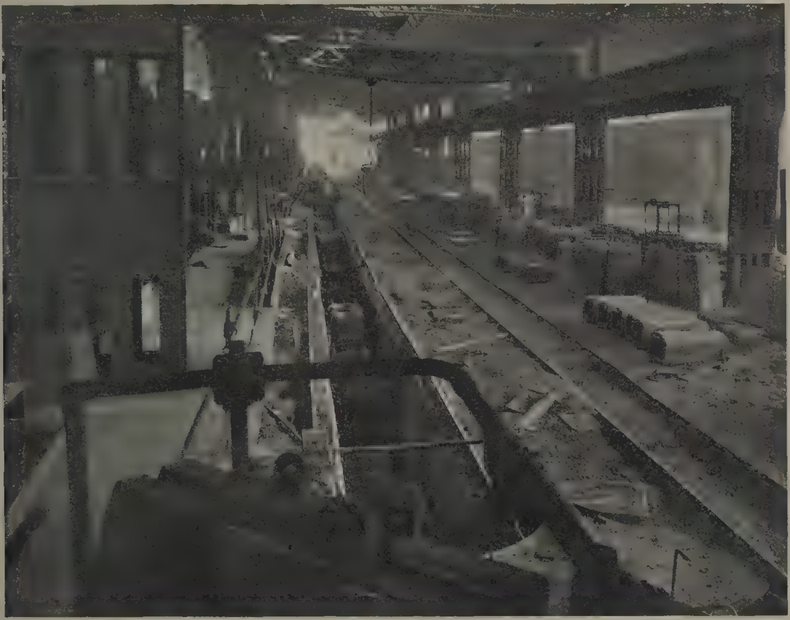

FIG. 16. CASTING PIT BAY OF SIEMENS FURNACE PLANT

Mechanical charging is in general use for furnaces above 30–40 tons capacity. Charging machines, electrically driven, may be either ground or overhead type. The steel making materials, previously weighed out in the assembling bay, are packed in specially designed boxes, and either hoisted on to the stage by cranes or run up on bogies. The machines are so designed that they can pick up a box, slew round, push it into the furnace, empty the contents, and replace it on the bogie. This obviously results in a considerable saving of time. Fig. 15 shows the working stage of a Siemens furnace with charging machine in position.

On many plants, particularly those in which the Talbot process is also worked, direct metal is charged. Iron from the blast furnace is taken to the melting shop and poured into a large capacity mixer, from which it is tapped off as required, and charged into the open hearth furnaces by means of a ladle slung from an overhead crane.

Melting. Charging completed, the furnace is put on gas and melting carried on as quickly as possible. When the melt is *overhead*—all solid portions submerged in molten metal—unmelted pieces must be kept loose from the bottom to prevent them eating into the hearth. The atmosphere of the furnace is an oxidizer and, therefore, the formation of a protective covering of slag must be carefully watched. If the charge does not appear to be making sufficient a few shovelfuls from a previous heat are thrown in. In about three hours the bath should be clear melted, by which time a partial removal of manganese, silicon, and carbon will have been effected. Thus far the oxidation is relative, and assuming that the 50 : 50 charge, quoted earlier, is being worked, the approximate composition of the *clear melted* bath would be: C 1·8, Si 0·90, Mn 0·25 per cent.

The Boil. The next phase is one of selective oxidation whereby the remaining silicon, manganese, and as much of the carbon as desired are removed. The reactions are closely analogous to those in the Bessemer converter with the important difference that the oxygen is provided in "solid" form—iron ore (Fe_2O_3). Ore is accordingly added as fast as the slag will take it although, at best, this can only be done gradually. The manganese and silicon, combining with the oxygen, form manganous oxide (MnO) and silica (SiO_2) respectively, and pass into the slag.

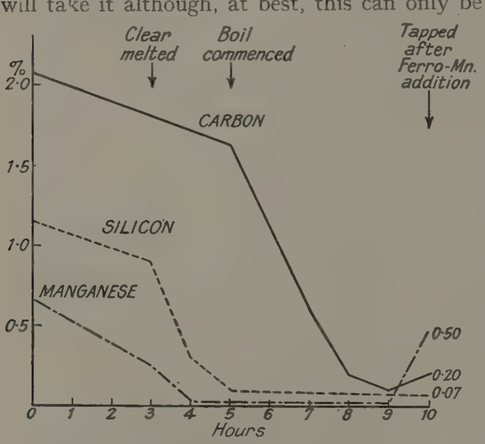

Fig. 17. Diagrammatic Representation of Removal of Principal Elements in Acid Siemens Charge

In metallurgical processes slags are the silicates formed by the combination of silica with alkaline earths and metallic oxides. As the percentage of silica increases, so does the viscosity of the fluid slag and, in furnace parlance, the sequence of the terms "thin," "thickening," "thick," and "off the boil" indicate this increase. The two important constituents of acid open hearth slag are silica and iron oxide. When the charge is clear melted the slag usually contains about 45 per cent SiO_2, and with the bath nicely on the boil this figure is remarkably constant at 49 per cent. To quote Longmuir,[1] "the purpose of the iron ore added is to maintain the slag in such a condition as regards its content of oxide of iron that it will act vigorously on the metal below, removing from it carbon, silicon, and manganese."

Until practically all the silicon has been oxidized the bath will not come on the boil and, for a good boil, two other factors are of vital importance—a sufficiently high temperature and a correctly constituted slag. The boil, of course, denotes the decarburization of the bath, and feeds of ore are made as the slag shows signs of thickening. The carbon content drops steadily, and the rate of fall is systematically checked and tabulated by rapid "colour carbon" analyses of spoon samples taken frequently and regularly. On the information thus obtained operations can be stopped at any desired point, when the bath is got into condition and tapped. If, for any reason, the carbon is found to have fallen too low, rectification can be made by *pigging back*. A sufficient weight of pig is charged to increase the percentage slightly above that desired, and matters proceed as before. Should the carbon "hang up"—the rate of fall be arrested—this can be put right by "rabbling" or lime additions. The rabbling operation consists of stirring the bath with a long steel bar, whilst lime has the effect of liberating small amounts of FeO from the slag and thus supplying a limited amount of oxygen.

Finishing. When nearing the requisite carbon percentage the slag should be allowed to thicken so that the fall is steadied and, with good practice no ore should be necessary for at least an hour before tapping. Wild and fiery steel will result from tapping a heat too soon after an addition of ore. Deoxidizers, such as ferro-silicon, ferro-manganese, and aluminium are necessary to ensure soundness of the finished steel, and as both of the first two contain carbon, allowance must obviously be made for this addition. If charged into the furnace these agents are broken up and preheated to redness in a small sand furnace, but may alternatively be fed into the stream of steel during tapping. Aluminium is added to the ingots by throwing a small piece into each mould as it is filled.

Tapping is carried out by picking away the clay filling of the tap hole by means of a pointed bar. If it is too hard then the bar is knocked through with a sledge hammer and withdrawn by means of blows upon a steel wedge which fastens a strong steel ring previously placed round the pricker bar.

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[1] *Elementary Practical Metallurgy* (Longmans, Green & Co.).

ELECTRO-TECHNICS FOR MECHANICAL STUDENTS

By A. P. Young, O.B.E., M.I.E.E.

LESSON XII

THE MAGNETIZATION OF IRON

In dealing with the magnetic effects of an electric current in Lesson VII, it was shown that a coil of wire carrying an electric current sets up a magnetic field, and that a piece of iron inserted in the coil behaves in exactly the same manner as a permanent magnet. In other words, the iron is magnetized through the agency of the current flowing in the *magnetizing coil.*

To create a flow of magnetic lines in a magnetic circuit, it is necessary to have a *magnetomotive force*, and it is the function of the magnetizing coil to create this force. The number of magnetic lines induced will be inversely proportional to the resistance or *reluctance* of the circuit, and the relationship between the three factors involved in a magnetic circuit is very similar to that for the electric circuit, thus—

$$\text{Magnetic flux } (\Phi) = \frac{\text{magnetomotive force } (M)}{\text{reluctance } (Z)} \quad . \quad (42)$$

The magnetomotive force (M) is solely dependent upon the product of current (I) flowing in the magnetizing coil, and the number of turns (S). This product (IS) is termed the *ampere-turns* of the coil. The magnetic resistance or reluctance (Z) is dependent on the dimensions and nature of the magnetic path.

1. Magnetomotive Force of a Coil. The magnetomotive force of a coil as deduced from first principles is given by the formula—

$$M = \frac{4\pi}{10} IS \quad . \quad . \quad . \quad . \quad (43)$$

where M = magnetomotive force
I = current in amperes
S = number of turns in the coil
(IS) = ampere-turns of the coil.

2. Reluctance of a Magnetic Circuit. The reluctance of a magnetic circuit, of uniform cross-section and comprising the same medium, is proportional to the length, and inversely proportional to the cross-sectional area. There is, however, another factor involved in a magnetic circuit, and this is the *permeability*, or magnetic conductance, of the medium through which the magnetic lines flow. It has previously been pointed out that magnetic materials, such as iron, cobalt, and nickel, offer considerably less resistance to the flow of magnetic lines than do non-magnetic mediums, such as air, and non-ferrous materials. In the case of the magnetic materials, the permeability (μ) varies with the flux density, and is greatly dependent upon the composition and structure of the material. On the other hand, the permeability of non-magnetic materials—like that of air, which is taken as the standard—is unity for all flux densities.

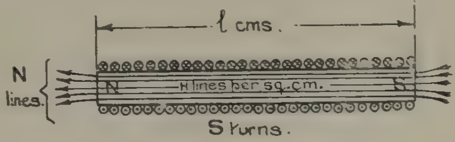


Fig. 89. A Solenoid

For a magnetic circuit of uniform section and having the same medium, the reluctance is determined from the formula—

$$Z = \frac{l}{A\mu} \quad . \quad . \quad . \quad (44)$$

Where, l = length of circuit in cms.
A = cross-sectional area in sq. cms.
μ = permeability.

Generally, a magnetic circuit consists of different mediums of varying dimensions. The determination of the flux in a composite magnetic circuit is a relatively simple matter, since the total reluctance is the sum of the separate reluctances of each portion. The flux is given by the expression—

$$\text{Flux} = \Phi = \frac{\frac{4\pi}{10} IS}{\frac{l}{A\mu} + \frac{l_1}{A_1\mu_1} + \frac{l_2}{A_2\mu_2}} \quad . \quad (45)$$

The flux density (B) in lines per sq. cm. for each portion of the circuit will be $\frac{\Phi}{A}$, $\frac{\Phi}{A_1}$, and $\frac{\Phi}{A_2}$ respectively.

3. The Magnetizing Effect of a Solenoid. A solenoid is a coil at least twenty times its diameter in length, and the flux induced in it, when the medium is air, can be accurately calculated. Such a coil is shown in Fig. 89, and consists of S *turns* of insulated copper wire wound on a long tube of non-magnetic material. It can be proved that the reluctance of the complete magnetic circuit is equal to $\frac{l}{A}$ so that by bending the coil into ring form as shown in Fig. 90, the flux threading the coil will remain unchanged.

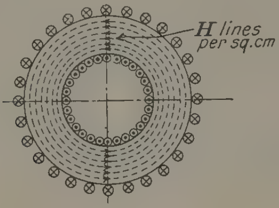


FIG. 90

In either case, the flux may be expressed as—

$$\text{Flux} = \Phi = \frac{\frac{4\pi}{10} IS}{l/A} = \frac{4\pi}{10} \cdot \frac{IS}{l} A$$

or,

$$\frac{\Phi}{A} = H = \frac{4\pi}{10}\frac{IS}{l} \quad . \quad . \quad (46)$$

The flux density in the coil is clearly H, and the length (l) for the ring form will be the *mean length* of the magnetic path. The quantity H, in lines per sq. cm., is known as the magnetizing force exerted by the coil. If instead of air, we have an iron core in the solenoid, then the flux density (B) will be very much greater, since iron provides an easier path to the flow of the magnetic lines. The ratio B/H is the permeability of the iron for that particular magnetizing force H. We thus have

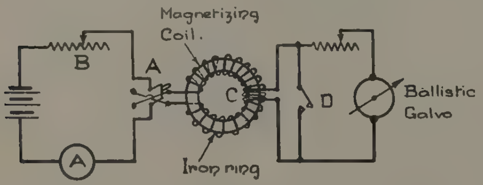


FIG. 91. CONNECTIONS FOR MAKING A PERMEABILITY TEST

$$\mu = B/H \quad . \quad . \quad (47)$$

or,

$$B = \mu \times H.$$

4. The Magnetization of Iron. The permeability of any material may be readily determined from equation (47) if we have means whereby we can measure the flux density (B) corresponding to a given magnetizing force H. For the purpose of making a test of this kind, the apparatus illustrated in Fig. 91 is employed. A ring of the material to be tested is uniformly wound with a magnetizing coil connected to a battery, through a reversing switch, which enables the current in the coil to be reversed and, in like manner, the flux in the iron ring. A rheostat (B) is inserted in the circuit to control the current, the value of which is indicated by the ammeter (A). The number of magnetic lines in the iron ring is measured by means of a "search coil" (C), wound on the iron ring underneath the magnetizing coil, and comprising a few turns (about 50) of No. 42 or 44 S.W.G. wire. This coil is connected to a ballistic galvanometer in series with a resistance, which is usually a plug resistance box.

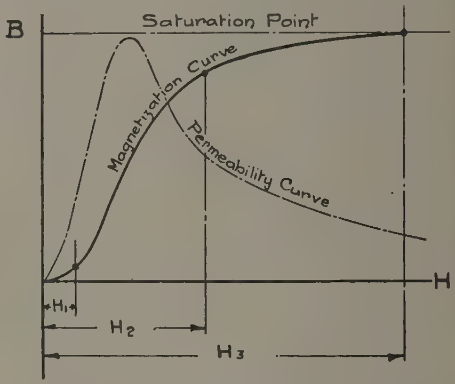


FIG. 92. A MAGNETIZATION CURVE

The *modus operandi* is as follows—

(*a*) Adjust the current in the magnetizing coil to some definite value I.

(*b*) Reverse the switch (A) several times in quick succession to bring the flux in the iron ring to a stable condition corresponding to a particular value of H as determined by equation (46). During this operation, the "shorting" switch (D), in the galvanometer circuit, should be held closed to ensure that the galvanometer is not disturbed.

(*c*) Next open switch (D) and quickly throw over the reversing switch (A). The galvanometer will then give a momentary deflection, which we will call d.

Now the ballistic galvanometer measures the *quantity* of electricity discharged through it, when the flux in the iron ring is reversed, and it can be shown that this quantity is given by the equation

$$Q = \frac{2S_1\Phi}{10^8 R} \text{ coulombs} \qquad . \quad . \quad (48)$$

Where, S_1 = number of turns on the search coil C
Φ = flux in the iron ring before and after reversal
R = total resistance of the galvanometer circuit.

The coulombs per division on the galvanometer scale can be determined by calibration, and if we denote this quantity by K, then the coulombs Q corresponding to a deflection d will be Kd. Equation (48) can now be written—

$$Kd = \frac{2S_1\ \Phi}{10^8 R}$$

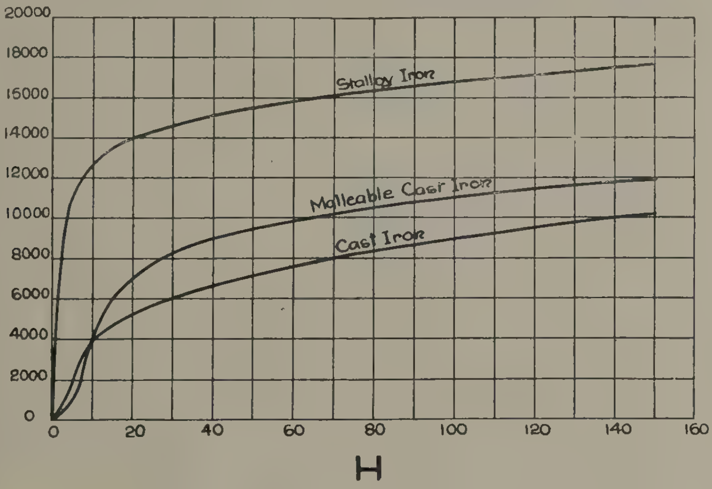


FIG. 93. TYPICAL MAGNETIZATION CURVES

and the value of the flux is then

$$\Phi = \left\{\frac{KR \times 10^8}{2S_1}\right\} \times d \text{ lines} \qquad . \quad (49)$$

It will be clear from this that the flux in the iron ring is *directly proportional to the deflection d*, and as all the factors within the brackets are known, the flux corresponding to any value of H can be readily calculated.

The cycle of operations (*a*), (*b*), and (*c*) is repeated for increasing values of H until any further increase in H results in little or no increase in flux, as indicated by the galvanometer deflection. For any given value of current the value of H will be—

$$H = \frac{4\pi}{10}\,\frac{IS}{l}$$

and the flux density B will be

$$B = \Phi / A$$

where, Φ = flux as determined from equation (49),
and, A = cross-section of the iron ring in sq. cms.

By plotting the values of H as abscissa, and the corresponding values of B as ordinate, we obtain a magnetization curve such as that shown in Fig. 92, which is typical of all grades of iron. Reference to this curve will show that magnetization curves for iron may be divided into three parts, viz.—

(*a*) In which the flux density B *increases slowly* with increase in magnetizing force. This applies for small values of H up to H_1.

(*b*) That portion in which the rate of increase in flux density B for increasing values of H is *very rapid*, corresponding to magnetizing forces from H_1 to H_2.

(*c*) The last portion of the curve between values of H_2 and H_3 in which the rate of increase in B is extremely *gradual* even for large increases in magnetizing force. The point corresponding to H_3 is called the *saturation* *point*, and for ordinary dynamo iron, H_3 is about 100.

The dotted curve shown in Fig. 92 is the "permeability curve," and is obtained by plotting the ratio $B/H = \mu$, as calculated for different points on the curve, against corresponding values of H. For dynamo sheet iron, the permeability attains a maximum of about 3,900 at quite a low value of H, after which it falls off with increase in magnetizing force, and at a H value of 100, corresponding to the saturation point, the value of μ is of the order of 180. Typical magnetization curves for samples of stalloy, malleable, and cast iron are given in Fig. 93.

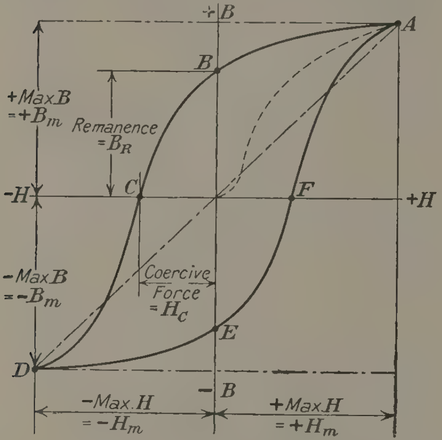


Fig. 94. Hysteresis Loops

5. Hysteresis. Whenever steel or iron is magnetized, it will be found to possess *residual magnetism*, after the magnetizing force is reduced to zero. Expressed another way, the curves connecting B and H for increasing and decreasing values of H are different. The residual magnetism for steel is much greater than that of soft iron, but even for soft iron the ascending B–H curves are different.

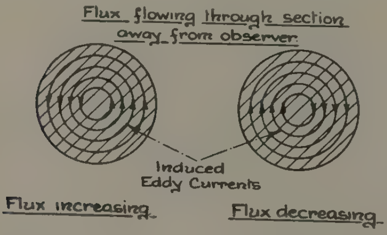


Fig. 95

With apparatus as shown in Fig. 91, it is possible to determine the values of B corresponding to increasing and decreasing values of H, when a piece of iron or steel is taken through a complete cycle of magnetization. Plotting the values of B against corresponding values of H for the various stages in the cycle of magnetization, curves as shown in Fig. 94 are obtained. Briefly, the stages are as follows—

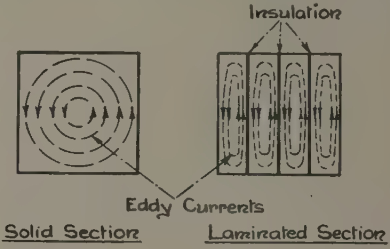


Fig. 96

(*a*) Starting from zero and increasing the magnetizing force to $+ H$, the B–H curve is that indicated by the dotted line.

(*b*) Gradually reducing the magnetizing force, the flux density falls along curve AB from the maximum value B_m to the value B, termed the *remanence*, and denoted by Br.

(*c*) To still further reduce the flux density, the current in the magnetizing coil is reversed, in order to apply a demagnetizing force, or a *negative* H. Increasing values of $-H$ reduce the flux density along curve BC until, corresponding to a demagnetizing force Hc called the *coercive force*, the flux density is again zero.

(*d*) Further increase in $-H$ results in an increasing flux density of opposite polarity, along curve CD, to a maximum $-B_m$.

(*e*) Reducing $-H$, the flux density falls along DE to a value E, when the magnetizing force is again zero.

(*f*) The current in the magnetizing coil is again reversed to flow in the original direction, in order to reduce the flux density to zero at F.

(*g*) Completing the cycle, the magnetizing force $+ H$ is increased until the flux density is again a maximum at A corresponding to $+ B_m$.

In plotting the curves for stages (*b*) to (*g*) inclusive, a closed loop $ABCDEFA$, called the *hysteresis loop*, is obtained. This loop is a measure of the energy dissipated as molecular friction and appearing as heat, when iron or steel is taken through a complete cycle of magnetization. This energy loss can be proved mathematically to be *exactly proportional to the area of the hysteresis loop*. When B is expressed as lines per sq. cm., the energy loss per cycle per cu. cm. of iron is given by the equation

$$\text{Hysteresis loss} = \frac{\text{area of hysteresis loop}}{4\pi} \text{ ergs.}$$

In commercial calculations, it is usual to express the hysteresis loss in watts. If the maximum flux density attained for each cycle, the cycles per second, and the total weight of material be known, the calculation of the loss in watts is a relatively easy matter.

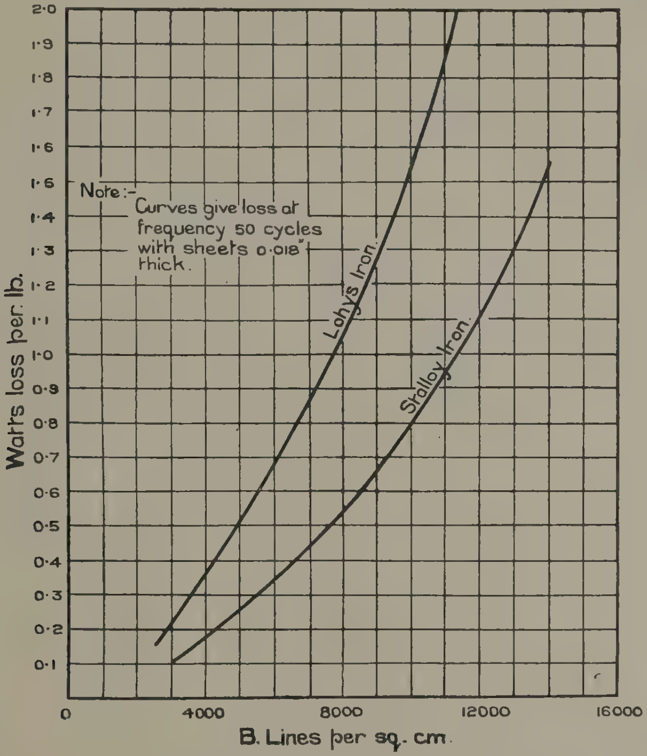


FIG. 97. TOTAL IRON-LOSS CURVES

The hysteresis loss in large alternators and transformers may be many kilowatts, and considerable attention has therefore been given to the manufacture of special grades of sheet iron in order to reduce the hysteresis loss. One of the most notable alloys is *stalloy*, invented by Sir Robert Hadfield, and manufactured by Joseph Sankey & Son, Bilston. It is an alloy of iron containing 3·4 per cent silicon, and has the advantage of low hysteresis loss and high specific resistance (ρ), whilst the permeability is little inferior to that of soft dynamo sheet iron.

The hysteresis loss per pound of iron per cycle for any particular grade of iron increases with increase in the maximum flux density (B_m), in accordance with a law expressed as follows—

$$\text{Hysteresis loss} = n.B^{k}_{m}$$

where, n = hysteresis constant
B_m = maximum value of flux density
k = a constant ranging from 1·5 to 2·0.

For general application Steinmetz's law $n \cdot B^{1\cdot6}_{m}$ may be used, although it is not always safe to assume that k is 1·6. For very low values of B_m the constant k is probably greater than 2.

6. Eddy Currents in Iron Circuits. The deleterious effects of eddy currents in iron have previously been referred to in Lesson VIII. It follows from Lenz's law that eddy currents will circulate in any mass of iron, in which the lines of force are changing, and the direction of these currents will be such as to oppose the change of flux that gives rise to them. The diagram in Fig. 95 will serve to illustrate this point.

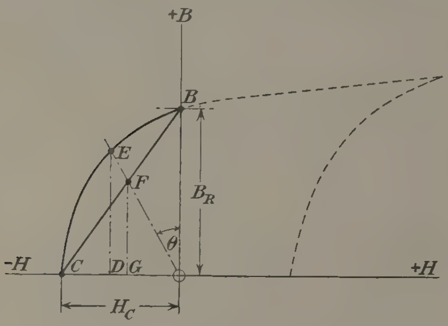


Fig. 98

Eddy currents produced by reaction cause *a retardation in the rate of change of flux*, and also cause *a dissipation of energy which reappears as heat.* These effects are overcome by laminating the iron circuit in which rapid changes in flux occur. The purpose of laminating the iron circuit is to constrain the eddy currents to separate paths in separate laminations, as illustrated in Fig. 96, thereby reducing their magnitude and effect. For given conditions, *the eddy current loss in watts, varies directly as the square of the thickness of each lamination and inversely as the specific resistance* (ρ) *of the material.* Also, the loss varies as the square of the flux density.

7. Energy Losses Due to Cyclic Magnetization. In iron circuits subjected to cyclic magnetization, the energy loss is the sum of the hysteresis and eddy current losses, and is termed the total iron loss. Total iron loss curves for two grades of iron, reproduced from a publication issued by Joseph Sankey & Son, are given in Fig. 97.

The watts loss per pound is very nearly proportional to B^2 as shown by the following equation—

$$\text{Total watts loss} = n \cdot B^{1.6} + kB^2$$

where, n = hysteresis constant

k = eddy current constant.

The following table giving the relative percentages for hysteresis and eddy current losses for stalloy iron at frequencies of 25, 50, and 100 cycles per second, shows that at moderate frequencies, k is always much less than n—

Flux Density B Lines/Sq. Cm.	Hysteresis Loss			Eddy Current Loss		
	25	50	100	25	50	100
	%	%	%	%	%	%
5,000	90	83	71	10	17	29
10,000	89	80	67	11	20	33
15,000	88	78	64	12	22	36

PERMANENT MAGNET STEEL

Permanent magnet steels are employed in the manufacture of many types of electrical apparatus, examples of which are, electric meters, telephone generators, electrical tachometers, small generators for motor-cycles, and magnetos for ignition purposes on internal combustion engines.

1. Criteria of Magnetic Quality. In general, the ability of a permanent steel magnet to provide a given flux under normal working conditions, is dependent on the shape of that portion of the hysteresis loop shown by BC in Fig. 98, since the area OBC is a measure of the energy stored in the magnet. The remanent flux density in a magnet after the magnetizing force has been reduced to zero is represented by B_R, and the value of the demagnetizing force H_C necessary to reduce the flux density to zero is the coercive force or retentivity of the magnet.

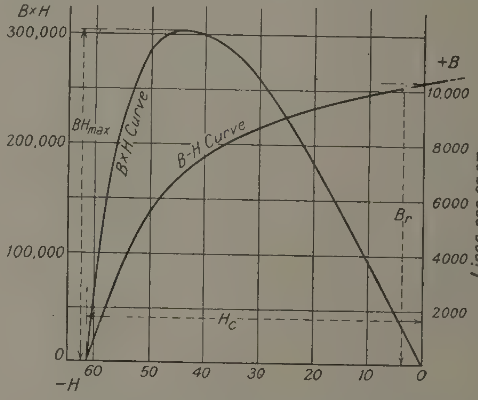


Fig. 99. Typical B–H and B × H Curves for Tungsten Magnet Steel

Knowing the shape of the B–H curve BC, the approximate flux density in the magnet can be determined by drawing a line OE, such that $\tan \theta$ is equal to the total reluctance included in the magnetic circuit. For example, the total reluctance in a well-designed magneto employing tungsten steel magnets, corresponds to a demagnetizing H of about 35. This is represented by OD in Fig. 98, and the working flux density corresponding to this reluctance is indicated by DE.

If we have two magnets having B–H curves as shown by BEC and BFC, the working flux densities will be DE and GF respectively. Now, whilst these two magnets have the same values of B_R and H_C, it will be seen that the flux density under working conditions of that represented by BEC is greater than that of magnet BFC. Clearly then it is necessary to have some indication of the shape of the B–H curve as well as the values of B_R and H_C, and this is best shown by a $(B \times H)$ curve, obtained by multiplying successive values of the demagnetizing H by the corresponding values of the flux density. Such a curve is shown in Fig. 99, giving typical magnetic characteristics for a tungsten steel magnet. It is now universally agreed that the true criterion of quality of a permanent steel magnet is the maximum product BH_{max}.

The steel most extensively employed in this country for permanent magnets is one containing 5 to 6 per cent tungsten. This steel has a remanence of about 10,000 and an average coercive force of 60, whilst the $(BH_{max.})$ value varies between 260,000 and 320,000. Steels containing about 2 per cent of chromium which have slightly poorer characteristics than tungsten steel, are also largely used.

2. Magnet Steels. In 1920, patents taken out by a Japanese firm disclosed the composition of a steel containing 35 per cent cobalt, and much smaller percentages of either molybdenum, chromium or tungsten, for which remarkably high magnetic characteristics were claimed. This steel has since been found to be capable of giving a coercive force of 250, a remanence of about 9,000, and a $(BH_{max.})$ of 1,000,000, which is roughly three times the $(BH_{max.})$ value of tungsten steel. This means that the length and weight of a 35 per cent cobalt steel magnet would be one-third that of a tungsten steel magnet for a given application. It is not surprising, therefore, that cobalt steel is finding increasing applications for all types of electrical apparatus, and especially on magnetos for aircraft, where weight is of paramount importance. Whether it will ultimately displace tungsten and chromium steels will very largely depend upon the prevailing price of cobalt, which at the moment does not permit of its widespread application. Other grades of cobalt steel, containing 15 and 9 per cent of cobalt, and having $(BH_{max.})$ values of the order of 620,000 and 450,000 respectively, are now employed for magnetos and small motor-cycle lighting generators, an example of the latter being shown in Fig. 100.

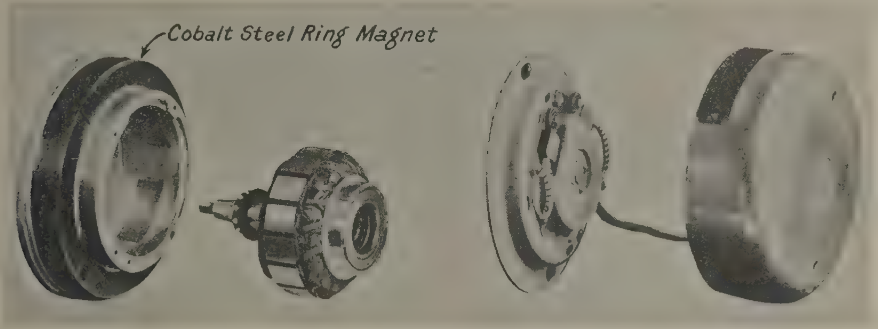


FIG. 100. COMPONENTS OF A SMALL MOTOR-CYCLE LIGHTING GENERATOR

THE BALANCING OF ENGINES

By E. B. COLE, B.Sc.

LESSON I

THE BALANCING OF REVOLVING MASSES

FROM his knowledge of Dynamics the student should already be acquainted with the meaning of *centrifugal force*, but it will be as well to recapitulate briefly the results.

When a body of weight W lb. revolves in a circle of radius r ft. with uniform angular velocity ω radians per second, then the body is subjected to an acceleration $\omega^2 r$ ft. per second per second radially towards the centre of the circle.

In other words, in order to keep the body moving in a circle a force of magnitude

$$\frac{W}{g} \times \omega^2 r \text{ lb.-wt.}$$

must act *upon* it, and this force is called the *centripetal force*, since it acts *radially inwards*.

If, however, the body is connected to the axis of revolution by a rigid or flexible link, this link will be in tension and exert on the body the required centripetal force inwards. The body, of necessity, exerts on the restraining link an equal and opposite force called the *centrifugal force* since it acts *radially outwards*.

Thus, any shaft which carries loads, whose centres of gravity do not coincide with the axis of rotation, e.g. crank webs, etc., will, when revolving, be subjected to centrifugal forces which will tend to set up vibrations and cause increased pressures on the bearings, resulting in wasted energy and unnecessary wear and tear.

Such forces are called *Dynamic Loads* on the shaft, and it is obvious that in the design of any system we must endeavour to arrange these loads such that the centrifugal forces exerted may form a *system in equilibrium*, thus mutually neutralizing their effects on the shaft. In other words, the shaft is *balanced* as regards revolving masses.

A short digression will here be made to discuss *vector quantities* and their addition and subtraction, since a simple semi-graphical method of dealing with balancing problems involving their use, which was developed by Professor Dalby, will here be followed.

A *vector quantity*, then, is one, which, in order to be completely specified, must be known both in *magnitude and direction*, the latter including *sense* of action.

Force, velocity, acceleration, couples, are examples, it being of no value merely to state the magnitude of such quantities; direction also must be given.

Such quantities may obviously be represented by straight lines, drawn parallel to the directions of, and of lengths, to scale, equal to the magnitudes of the respective quantities.

These lines are termed *vectors*, and an arrow head indicates their sense.

It is assumed that the reader has studied the elements of Statics and is familiar with the "polygon of forces," by which we are enabled to determine graphically the *resultant* of any number of forces acting at a point in one plane.

All we really do is to draw the various vectors representing the forces, "add" them together and find their *vector sum*, i.e. their *resultant*.

What is true for forces is equally true for any vector quantity, and therefore by means of a *vector polygon* we may find, graphically, the vector sum or resultant of any such quantities.

In balancing problems we shall be dealing with centrifugal forces acting radially outwards from a shaft, and also, as we shall see, with couples produced by these forces about any given point along the shaft, and it will be our object to determine the resultant force and resultant couple acting. This will be done by means of the "polygon of vectors"—the construction of the force polygon should need no explanation here, but in case the procedure is unfamiliar to the student a brief description is here given.

In Fig. 1 (*a*) the directions of three forces acting at a point in one plane are indicated as well as their magnitudes, i.e. 1, 3, and 2 lb. respectively.

The vector xy, Fig. 1 (*b*), represents the force of 2 lb. since it has been drawn parallel to the direction of this force, and its length has been made 2 units to scale. The arrow-head shows the sense in which the force acts.

To find the resultant of these three forces their vectors are "added" by placing them end to end, *the arrow-heads all following one another in one direction* as shown in Fig. 1 (*c*). The *closure* of the polygon so formed represents in magnitude and direction the resultant of these forces. The direction sense is in opposition to the arrow-heads of the force vectors as shown.

It does not matter in which *order* we take the vectors, as is shown by Fig. 1 (*d*), the resultant is always the same, but it is most important to see that the arrow-heads are in the right direction and taken in order.

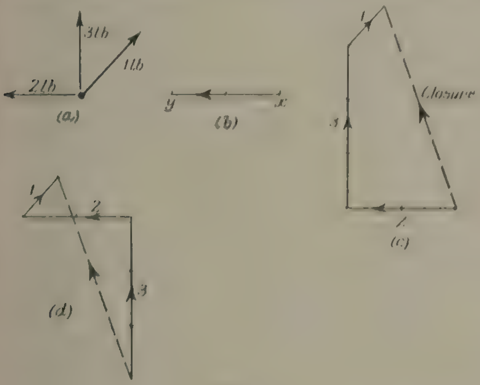


FIG. 1

If a polygon should close, this means that there is no resultant effect on the point at which the vector quantities act, or, in other words, these quantities neutralize one another.

Let us now take our first problem in balancing—

To balance any number of masses revolving in one plane of revolution perpendicular to the shaft or axis.

Suppose we have a series of masses of weights W_1, W_2, W_3 lb. etc., at radii r_1, r_2, r_3 . . . ft. respectively from the axis of revolution and all in the same plane, as indicated in end elevation in Fig. 2 (*a*), then the *resultant dynamic load* on the shaft will be the vector sum of all the centrifugal forces set up, i.e.

$$R = \text{vector sum } \frac{W_1}{g} r_1\omega^2 + \frac{W_2}{g} r_2\omega^2 + \ldots$$

$$\text{i.e.} \quad R = \frac{\omega^2}{g} \sum Wr \text{ lb. wt.,}$$

where Σ means "vector sum of."

By drawing a "polygon of forces" for this system we can obtain graphically the magnitude and direction of this resultant R.

Note that, as $\frac{\omega^2}{g}$ is a constant for all the forces, these are proportional to Wr and, in drawing the polygon, the sides need only be constructed representing Wr.

Such a polygon has been drawn in Fig. 2 (*b*) for the case shown.

The dotted line forming the closure represents in magnitude ΣWr, i.e. R, which acts in the direction shown.

To take a concrete example let

$W_1 = 7{\cdot}5$ lb., $r_1 = 2$ in.
$W_2 = 2$ lb., $r_2 = 3$ in.
$W_3 = 10$ lb., $r_3 = 1$ in.

Then $W_1r_1 = 15$
$W_2r_2 = 6$
$W_3r_3 = 10$

These values of Wr have been scaled off on the "force diagram," Fig. 2 (*b*), being taken in order in an anti-clockwise direction, and the closure of the polygon is found to be

$$\Sigma Wr = 16{\cdot}9.$$

Thus, at ω radians per second the dynamic load on shaft is

$$R = \frac{16{\cdot}9}{32 \times 12} \times \omega^2 = {\cdot}044\,\omega^2 \text{ lb. wt.}$$

(The values of r having been taken in inches.)

Balance of this system can at once be effected by arranging a weight W_0 at radius r_0, such that the centrifugal force it sets up is equal and opposite to R, i.e.

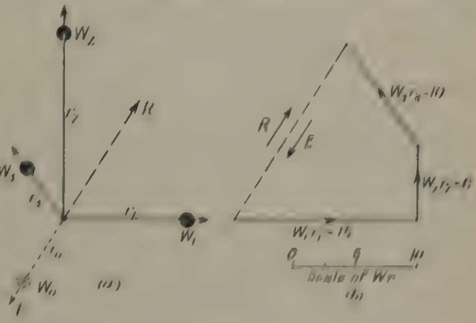


FIG. 2

$$-\frac{W_0}{g} r_0\omega^2 = R = \frac{\omega^2}{g} \sum Wr$$

or $$-W_0 r_0 = \sum Wr.$$

In other words $W_0 r_0 \cdot \frac{\omega^2}{g}$ is the *equilibrant E* of the forces and the force polygon now closes, there being no unbalanced force.

In the example considered, if we took $W_0 = 5$ lb., then r_0 would have to be $16{\cdot}9 \div 5 = 3{\cdot}38$ in.

The one condition necessary that any number of forces in one plane should be in balance is that

The force polygon must close, i.e. $\Sigma Wr = 0$.

To balance any number of masses revolving in separate parallel planes.

Consider first two masses arranged radially opposite one another on a shaft, such that

$$-W_1 r_1 = W_2 r_2$$

but in separate parallel planes distant x apart.

Fig. 3 gives a pictorial view of these conditions. Each weight sets up the same centrifugal force F on rotation, so that the force polygon closes as before, *but the shaft is not now in balance being subjected to a couple* $= Fx$.

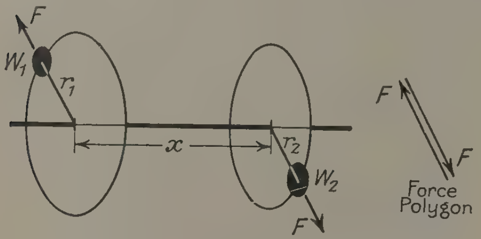


FIG. 3

We are thus faced with the further problem of dealing with couples as well as forces, and in order to visualize their effects, let us first of all consider the effect that a force F, acting on a shaft in any plane of revolution A, has in regard to any other parallel plane B.

In other words, if the shaft were imagined to be held only at the point B, what effect would a force F at A produce at B. The system is that of a cantilever supported at B under the action of a force F distant x from B.

Clearly then this force produces a *reaction* at the imaginary support B equal and parallel to itself, and a *moment or couple* about B of amount Fx. Fig. 4 illustrates this.

Hence, the effect of a force F, acting on a shaft in any plane of revolution A, on any other parallel plane B distant x from A, is to exert—

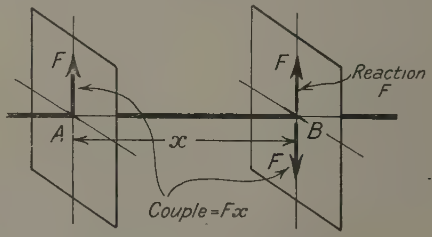
1. *An equal and parallel force F at B.*
2. *A couple perpendicular to the plane* $= Fx$.

Thus, if we have any number of masses revolving in different parallel planes, their effect on any given parallel plane of reference is similarly—

1. To introduce in this reference plane forces equal and parallel to the respective centrifugal forces acting. These forces may obviously be combined into one resultant $= \frac{\omega^2}{g} \sum Wr$.

Note that the *position* of the reference plane has no effect on the magnitude of the forces, nor, therefore, on their resultant.

2. To exert couples perpendicular to the plane, which, being vector quantities, may also be combined into one resultant couple $= \frac{\omega^2}{g} \sum Wr \, . \, x$.



Shaft imagined to be held at point B only

FIG. 4

Here the position of the reference plane does affect the value of the couples, and therefore of their resultant, since it determines the values of x.

It will be realized that the actual unbalanced effect produced by a series of given centrifugal forces on a shaft is a constant for any given conditions having nothing whatever to do with any particular plane, but when we refer these forces to a given plane and examine their effect on that particular plane, then the resultant couple they exert on that plane does depend on the position of the plane chosen.

Hence the effect of any such system may be reduced to *a single unbalanced force in any*

(*Continued on page* 1352)

PIONEERS OF ENGINEERING

By J. F. Corrigan, M.Sc., A.I.C.

LESSON XXV

A MODERN TREND—THE DIESEL ENGINE

DR. RUDOLF DIESEL (1857–1913)

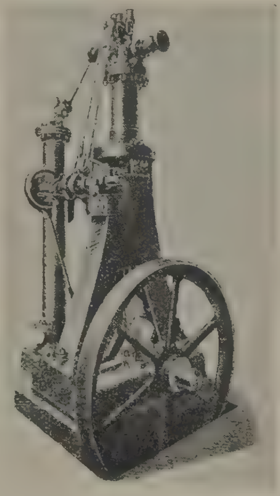
The Diesel engine is a peculiarly modern development of internal combustion engine design. Indeed, its inherent possibilities were scarcely recognized at the untimely death of its inventor, Dr. Rudolf Diesel, in 1913, but of more recent years, however, its capabilities have been brought well to the fore.

Dr. Rudolf Diesel

Diesel engines have the advantage of low fuel consumption. They are steady in action, and will run reliably over long periods. Their mode of action is not difficult to follow. On the downward stroke of the piston, air is drawn into the cylinder. During the following up stroke, the charge of air is subjected to a compression of about 500 lb. per sq. in. In accordance with a definite theoretical law, a rise of temperature accompanies this compression, so much so, that at the moment of maximum compression the temperature of the air is sufficiently high to ignite a charge of fuel which is injected into the cylinder. Spontaneous ignition, therefore, takes place, and the piston is driven downwards.

It must be noted that in the Diesel engine, the conditions of fuel supply are so regulated that there is little change of pressure during the burning of the fuel, the end aimed at being to cause combustion to take place steadily, without explosion, and at a sensibly constant pressure.

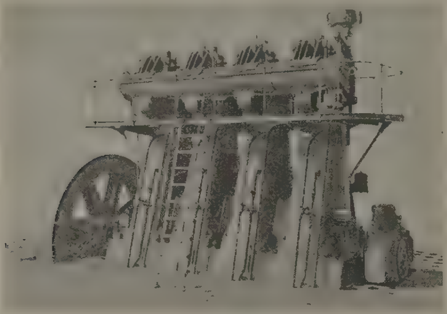
In an engine of the four-stroke Diesel type, the cycle of operations is (1) Suction of Air, (2) Compression of Air, (3) Power stroke, (4) Exhaust. The engine is started by means of air pressure from reservoirs of highly compressed air, and this supply of air also serves, in some cases, for the purpose of injecting the charge of fuel into the cylinder after each compression stroke of the piston.



Diesel's First Engine, 1892

The inventor, Rudolf Diesel, was born in Paris of German parents, in 1857. He studied at the technical colleges of Augsberg and Munich, and during his later years, he managed an engineering works in Paris.

The circumstances in which Diesel was led to invent his engine are of interest. About 1890, he published a book on the theory of heat engines, in which work he proposed in outline the principles of his engine. The volume attracted so much attention that the author was induced to give his whole attention to the subject. In this work he was financed by Messrs. Krupps, Sulzer Bros., and other engineering concerns.



A Vertical Four-cylinder Diesel Engine for Land Duty

(From E. Supino's *Land and Marine Diesel Engines* Griffin & Co., 1915)

The first experimental Diesel engine was constructed in 1893. It utilized coal dust as a fuel, and on one occasion its inventor nearly lost his life owing to a violent explosion which occurred during its working.

Diesel rapidly improved upon this early attempt, and within a number of years the engines began to be turned out in relatively large numbers. In 1910, the number of Diesel plants in regular operation in the United Kingdom could be counted upon the fingers of one hand. In 1920, however, no less than 120 commercial Diesel plants were in operation in this country, and their number has been recently still more increased.

Diesel met with a tragic end. Travelling to England on the night of 30th September, 1913, on the Antwerp-Harwich mail packet, he fell overboard and was drowned. He had financial worries, it is known, but almost certainly his death was not an intentional one on his part, as has been suggested. Thus ended the life of one of engineering's very latest pioneers. Diesel was man with one salient idea. He kept it before him all his life. He worked at it unceasingly. And now, fifteen years after his death, the results of his labour are beginning to bear abundant fruit, and to be of service in the furtherance of engineering, and also to the civilized community in general.

(Conclusion)

THE BALANCING OF ENGINES

(*Continued from page* 1350)

parallel plane of revolution, together with a couple perpendicular to this plane.

If this resultant force and resultant couple are both zero for any plane of reference the shaft is obviously in balance, being subjected to no disturbing effects.

The conditions for perfect balance are thus—

1. *There must be no resultant force acting on the shaft*, or the force polygon must close, i.e. $\Sigma Wr = 0$.

2. *There must be no resultant couple about any parallel plane*, i.e. the couple polygon drawn for any reference plane must close, or $\Sigma Wrx = 0$.

In the next lesson we shall explain the method of drawing the couple polygon and how balance is effected.

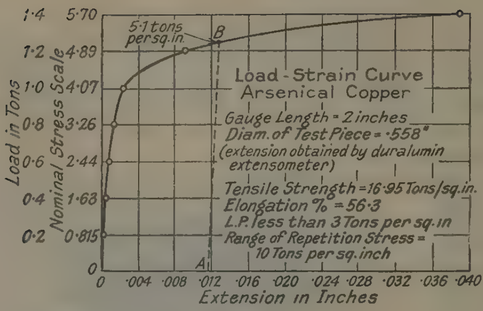


FIG. 1

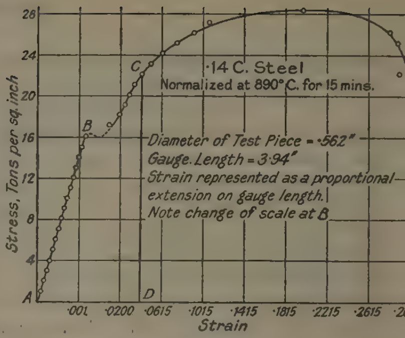


FIG. 2

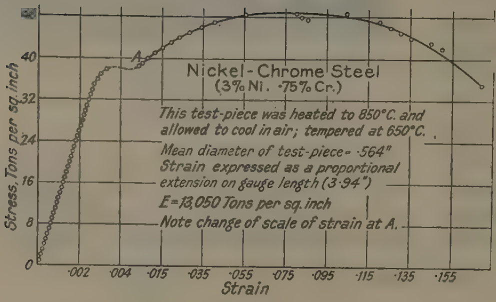


FIG. 3

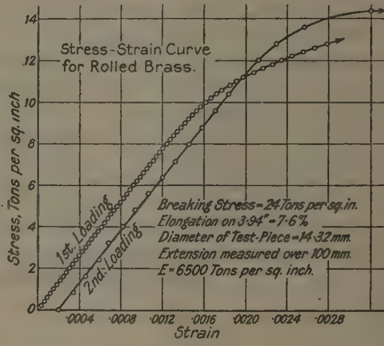


FIG. 4

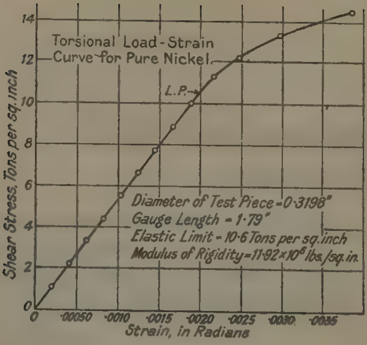


FIG. 5

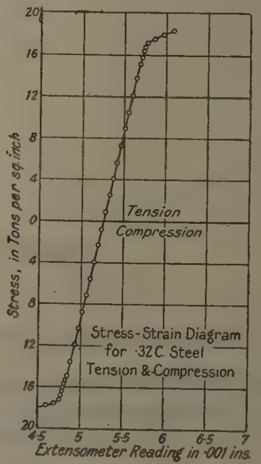


FIG. 6

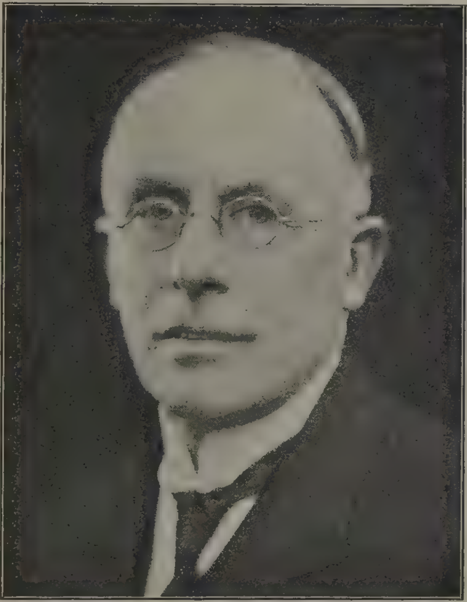
STRESS-STRAIN CURVES FOR VARIOUS MATERIALS

REPEATED STRESSES IN MATERIALS

By Professor F. C. Lea, D.Sc., M.I.Mech.E., M.Inst.C.E.

PART I

The effect of repeated stresses on materials is not only of great interest scientifically, but is also of great importance in connection with the design of machinery, tools of all types, and structures that may be subjected to repeated loads. During the last sixty years a great deal of data has been obtained from machines specially designed to investigate this problem, and from the great laboratory of experience, but up to the present it can hardly be said that a completely satisfactory theory of the mode or cause of failure has been worked out, and, furthermore, it can hardly be said as yet to which of the other properties of materials the safe range of repetition stress is related.



Professor F. C. Lea, D.Sc., M.Inst.C.E.

Professor of Engineering, Sheffield University; formerly Professor of Civil Engineering, University of Birmingham; has designed large reinforced concrete and other structures; awarded the Telford Prize (twice) and the Crampton Prize, etc.; Author of "Hydraulics"

Tensile, Compressive and Shear Failure of Materials. Whatever state of stress to which a piece of material is subjected, or whatever form of loading it has to resist, the resultant stress on any plane in the material can be resolved into a normal stress, which may be either a tension or compression, and a tangential or shear stress. An element of a machine, such as a crankshaft, may be subjected to bending moments, twisting moments, and direct loads, and during each revolution all these may pass through cycles which cause on any given plane in the shaft cycles of normal stress and shear stress. The conditions may be simple or complicated, but the final problem resolves itself into the question, what ranges of stress, normal stress and shear stress, can be allowed on any plane without risk of fracture?

The behaviour of materials under statically applied stresses can be easily determined in the special types of machines and apparatus devoted for this purpose.

Figs. 1 to 6 show curves obtained from tests on materials in simple tension, compression, and under pure shear produced by a twisting moment or torque. The first portion of the curve is either a straight line, or approximates to a straight line, and is known as the elastic portion of the curve. The point at which the curve deviates from the straight is generally called the elastic limit or, better still, the limit of proportionality. From the point of view of repeated stresses it is desirable to be clear, if possible, as to what is really implied by an elastic material. The ordinary type of extensometer used in connection with testing machines probably measures strains of the order of $\frac{1}{10000}$ to $\frac{1}{50000}$ with a reasonable degree of accuracy. Torsionmeters used in connection with shear tests also measure small angular strains with a considerable degree of precision. In both these types of tests, the "straight line" portion of the diagrams are straight within the accuracy of the instruments. More precise measurements show, however, that when the material is loaded through cycles of stress which are not of very great range, the "straight line" is not exactly straight, but that in going through the cycle a hysteresis loop is obtained.

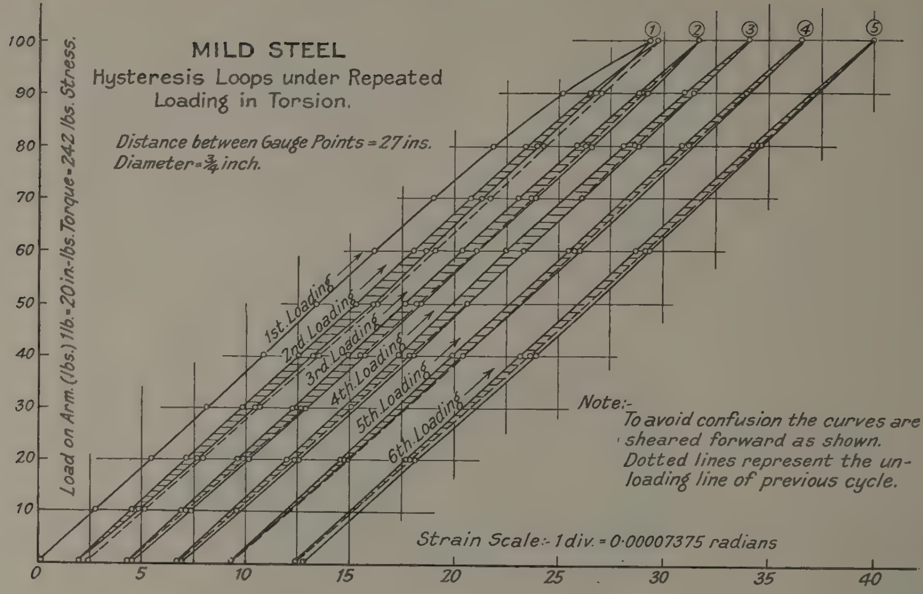

MILD STEEL
Hysteresis Loops under Repeated Loading in Torsion.
Distance between Gauge Points = 27 ins.
Diameter = 3/4 inch.
Load on Arm. (lbs.) 1lb. = 20 in.-lbs. Torque = 242 lbs. Stress.
100
90
80
70
60
50
40
30
20
10
①
②
③
④
⑤
1st. Loading
2nd. Loading
3rd. Loading
4th. Loading
5th. Loading
6th. Loading
Note:-
To avoid confusion the curves are sheared forward as shown.
Dotted lines represent the unloading line of previous cycle.
Strain Scale:- 1 div. = 0·00007375 radians
0
5
10
15
20
25
30
35
40

FIG. 7

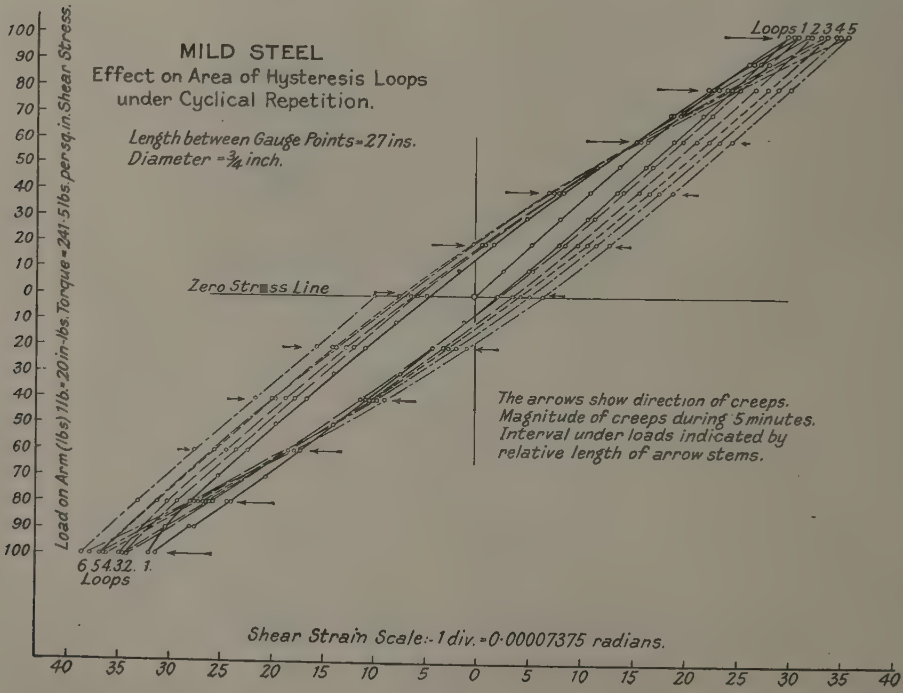

MILD STEEL
Effect on Area of Hysteresis Loops under Cyclical Repetition.
Length between Gauge Points = 27 ins.
Diameter = 3/4 inch.
Load on Arm (lbs) 1lb. = 20 in-lbs. Torque = 241·5 lbs. per sq. in. Shear Stress.
100
90
80
70
60
50
40
30
20
10
0
10
20
30
40
50
60
70
80
90
100
Loops 1 2 3 4 5
Zero Stress Line
6 5 4 3 2. 1.
Loops
The arrows show direction of creeps.
Magnitude of creeps during 5 minutes.
Interval under loads indicated by relative length of arrow stems.
Shear Strain Scale:- 1 div. = 0·00007375 radians.
40
35
30
25
20
15
10
5
0
5
10
15
20
25
30
35
40

FIG. 8

Such loops are shown in Figs. 7 and 8, obtained by the author from a mild steel bar in a special torsion machine and using an optical torsionmeter. This indicates that when the specimen is loaded slowly in pure torsion through the range shown in the cycle, work must be done on the specimen and internal heat generated. If the range of stress is below a certain amount the cycles close, or in other words, there is no permanent deformation of the material, providing that the cycle finishes at the point of commencement. If the cycles are repeated, and again the range of stress is below a certain amount, the areas of the hysteresis loops may diminish, whereas, if the range of stress is above a certain range, the hysteresis loops may increase in area. The curves of Fig. 7 were obtained from mild steel for a range of stress 0 to 10·4 tons per sq. in. The loops diminish in width as the cycles are repeated. When the range of stress was increased to ± 10·4 tons per sq. in., Fig. 8, the hysteresis loop increased in width and in area. Fig. 9 shows how the width of the loop and the stress at zero strain increased with the number of cycles. Anticipating for a moment what is to be said later, it can be stated that a specimen subjected to a range of stress 0 to 10·4 tons per sq. in. would withstand thousands of millions of repetitions without fracture, whereas a specimen subjected to a range of ± 10·4 tons per sq. in. would probably break after a comparatively small number of repetitions.

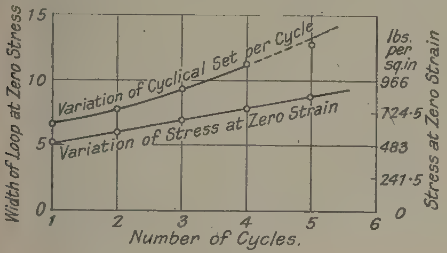


FIG. 9

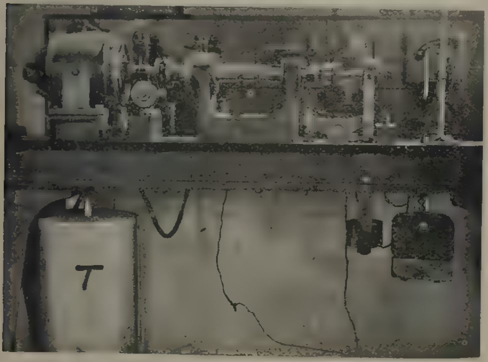


FIG. 10. THE 2,000 R.P.M. COMPOUND LEVER MACHINE

Hysteresis Loops and Generation of Heat. It would naturally be expected that, if when subjected to repeated stresses such hysteresis loops are produced, heat would be generated. Experiment shows very definitely that such is the case. By the use of thermocouples it is easy to show that a rise of temperature of the specimen may take place, but a rise of temperature does not indicate of necessity that fracture at the particular range will take place. With wrought iron, copper, nickel, and other substances, at particular ranges, spurts of heat may take place and then die away; indicating that hysteresis may be obtained and then gradually become zero.

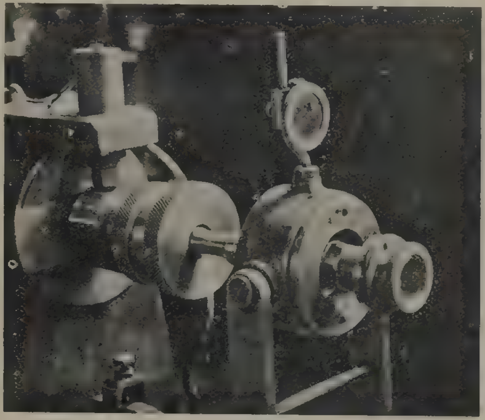


FIG. 11. VIEW OF ROTATING—ALSO STATIONARY MIRRORS

Just before the fracture, under repeated stresses, occurs, it is frequently observed that considerable heating of the specimen takes place, showing that after a certain number of repetitions, a condition arrives which allows of considerable hysteresis followed by fracture.

The question naturally arises, can the range of repetition stress be anticipated from the so-called elastic portion of load strain curves, such as those shown in Fig. 1? Unfortunately, in general, the answer is no. The author and others have shown very definitely that the length of the straight line portion of the load-strain or stress-strain curve does not indicate the safe range of repetition stress. In the case of normalized mild steel subjected to equal plus and minus stresses, i.e. with zero mean stress, the range agrees fairly well with the range of stress indicated by the total length of the straight portion of the curve obtained from tension and compression tests. Such a curve is shown in Fig. 6. The range of stress for this steel at this mean stress was found to be ± 15·2 tons per sq. in.

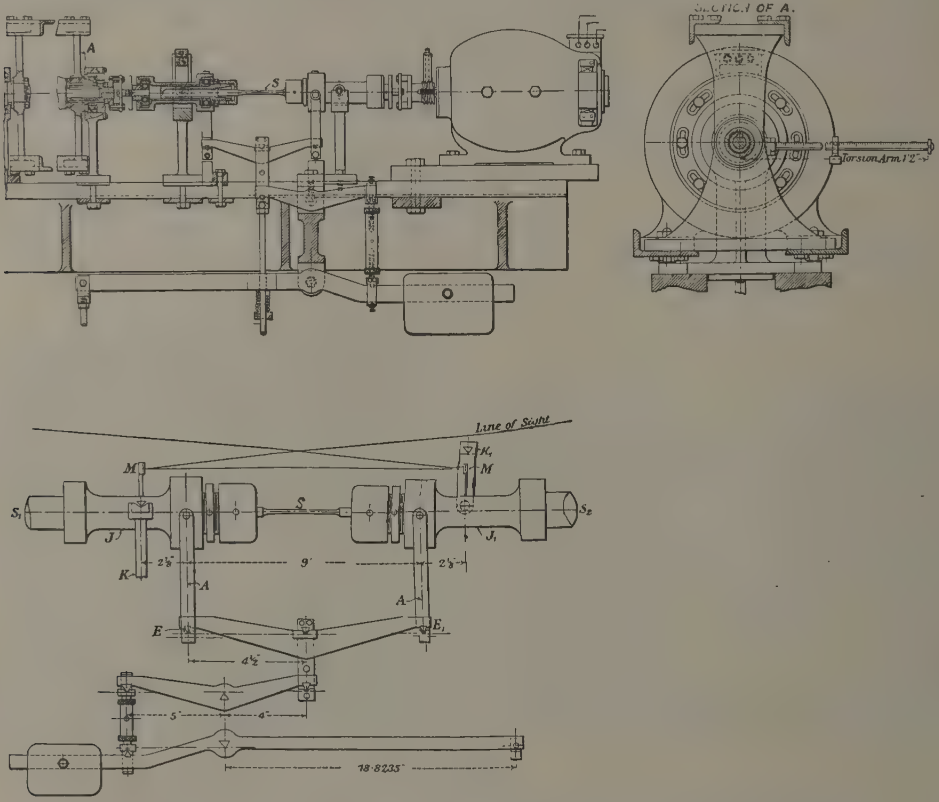


Fig. 12. Machine for Producing Constant Torque and Alternating Bending

Load-strain Curves Beyond the Limit of Proportionality. When a statical load in tension or compression is carried beyond the limit of proportionality, Figs. 1 to 4, a sudden yield may take place, Fig. 2, or the deviation may take place gradually, Fig. 1. If at a point *C*, Fig. 2, the load is released, the unloading curve is *CD*, and on reloading, the curve may be straight up to the point *C*, or near to *C*. It should be noted that on the scale of the portion *AB* of the curve, the line *CD* would be parallel to *AB*.

Under repeated stresses it is quite possible to load a specimen through a considerable range of stress, say from *D* to *C*, without fracture even after many millions of repetitions, but the total safe range is smaller than when the mean stress is zero. If for the maximum stress *C* load-strain curves are taken for various ranges of stress, it is found for mild steel that the safe range of stress agrees fairly well with that range of stress for which the load strain curve is straight, but the author's tests at high temperatures show again that this is not generally true.

It can be stated that for zero mean stress, the total range of stress for most materials, bears a more definite relationship to the ultimate tensile strength than to any other property, but the ratio of the safe range to the ultimate stress varies with the material, its history, and its heat treatment and, in general, the only way to determine the safe range of stress for any given mean stress is by direct experiment.

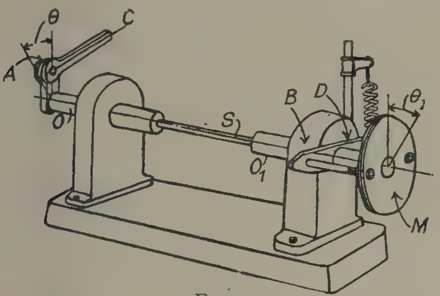


Fig. 13

Types of Machines for Experiments on the Effect of Repeated Stresses. Fig. 10 shows a simple type of Wöhler bending machine, designed and used by the author for tests on specimens subjected to cycles of bending stress. The specimen is a simple cantilever loaded through the ball bearing. The bending moment is varying along the specimen, but the specimen can be so shaped that over a given length the stresses are practically the same at each section of the length. The load can be hung directly from the bearing, or it can be applied by admitting water to the tank and through the lever system under the bed of the machine. The tank allows load to be applied or taken off very gradually. The tank can be suspended from a spring balance and the load at any instant read directly. On the end of the specimen is a mirror, see also Fig. 11, and load-strain curves can be obtained while the specimen is running.

Fig. 12 shows a second machine[1] designed and used by the author in which the specimen S is subjected to a constant bending moment, and it may also be subjected simultaneously to a constant torque. In this machine specimens can therefore be subjected to combined direct stress produced by bending and shear stress due to torsion.

Torsion Machines. Fig. 13 shows a diagram of a machine designed and used by the author for repetition stresses in torsion. The specimen S is fixed in two collets. One of the collets is fixed to a spindle, the mass M and the lever D.

The other is connected to a crank AO, oscillated by the connecting rod C.

To the arm BD is attached, at a definite radius, a spiral spring in which any initial tension can be set up by an adjusting screw and thus any initial torque in the specimen. On oscillating the crank, the mass M and arm D oscillate through a very small angle θ_1; this is kept small by so designing the specimen that a node occurs nearer to the right-hand end than the left. The oscillation of the arm being small the pull in the spring and thus the mean stress remain practically constant.

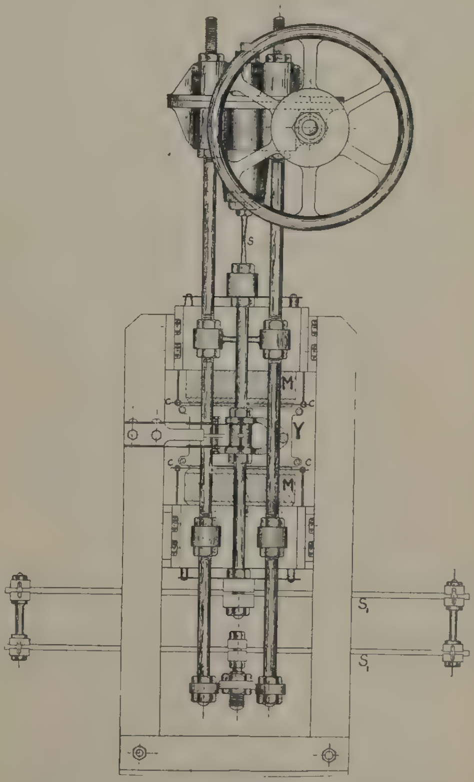


Fig. 14. Haigh Alternating Stress Machine

For description see *Engineering*, 26th August, 1926.

The angle θ_1 is measured as the specimen oscillates, by the movement of a spot of light over a scale on ground glass at the end of a dark box or camera. The torque, from which the stress in the specimen can be calculated, can be determined when the moment of inertia, of the oscillating mass, the angle θ_1, and the speed of the machine are known. A second type of machine used and designed by the author is similar to the one described, except that the end of the specimen is connected rigidly to and in series with an optical torsionmeter. Any initial torque can be put in the specimen, and then any amplitude θ can be imposed by the oscillating crank. The torque in the specimen is measured by the movement of the spot of light reflected from the middle of the torsionmeter on to a ground glass scale.

Haigh Machine. One of the most ingenious and useful machines for testing materials under repeated stress is that designed by Dr. Haigh, and made by Messrs. Brunton's of Musselburgh.

The specimen S, Fig. 14, is connected to a fixed crosshead and to a movable crosshead which is connected by rods to the wrought-iron yoke Y. This is connected also rigidly to the leaf springs S_1. The yoke is between two electro-magnets, M and M_1, which are magnetized by an alternate current at 2,000 cycles per minute. When the upper magnet is magnetized, a compressive force acts on the specimen, and when the lower is magnetized a tensile force acts upon the specimen. Any mean stress desired can be impressed upon the specimen, either tensile or compressive by putting initial strain upon the leaf springs in the required direction. The magnets then impose an additional range of stress on the specimen; this can be modified to any extent within the range of the machine. The pull or push exerted by the magnets can be measured by the voltage generated in coils arranged near to the air gaps between the electro-magnets, and the wrought-iron yoke Y, and is measured by a voltmeter. The voltage generated in these coils is proportional to the rate of change of the lines of force passing between the magnet and the yoke; the pull on the yoke is proportional to the lines of force, i.e. since the speed of the alternate current machine is kept constant, to the voltage generated.

Other types of machines have been developed by various experimenters, but these are either of the Haigh type giving direct stresses, such as the Smith Reynolds oscillating mass machine, of the Wöhler type giving bending stresses, of the torsion type giving shear stresses, or of the combined bending and torsion type such as shown in Fig. 12, or of the type designed by Mason (Proc. Inst. Mech. Engs., January–May, 1917).

(*To be continued*)

APPLIED MECHANICS

By Geo. W. Bird, Wh.Ex. B.Sc., A.M.I.Mech.E., A.M.I.E.E.

LESSON XXIV

MECHANICS OF FLUIDS

Venturi Meter

The Venturi meter consists essentially of a converging-diverging pipe inserted in the main pipe line with provision for measuring the difference of pressure heads at the full bore and at the throat sections. A diagram of the meter in section is shown in Fig. 151. At the full bore section a number of small holes serve to transmit the pressure to an annular ring from which a pipe may be taken to one end *A* of a glass tube containing mercury. A similar arrange- of small holes, etc., at the throat section transmits the pressure to the other end *B* of the glass tube, so that the difference of levels $a_2 b_1$ is an indication of the pressure difference existing between the full bore and throat sections. It is usual to place the axis of the meter horizontal. The diameters at the full bore and throat sections must be accurately known and, to reduce friction loss, the metal surfaces should be as smooth as possible. The mercury tube arrangement of Fig. 151 is, in practice, usually replaced by an automatic recording apparatus in which, by a combination of movements, controlled by clockwork, and a second movement due to the rise or fall of floats, a continuous record of the flow is obtained.

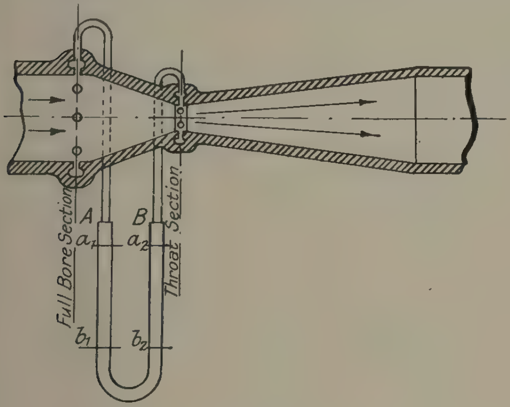


Fig. 151

Application of Bernoulli's Theorem to the Venturi Meter

Example. The full bore diameter of a Venturi meter is 3 ft. 6 in., and the throat diameter 1 ft. 3 in. The difference between the pressure heads at the full bore and throat sections is indicated by 15·5 in. difference of level in the mercury tube, Fig. 151, sp. gr. of mercury 13·56. Determine the flow of water through the meter in gallons per minute, the axis of the meter being horizontal.

$$\frac{p_1}{w}+h_1+\frac{v_1^2}{2g}=\frac{p_2}{w}+h_2+\frac{v_2^2}{2g} \qquad . \quad (1)$$

Where p_1 = pressure in lb. per sq. ft. at full bore.
v_1 = velocity in ft. per second through full bore section.
p_2 = pressure in lb. per sq. ft. at the throat section.
v_2 = velocity in ft. per second through the throat section.
h_1 = head at full bore section above or below some agreed datum level.
h_2 = head at throat section above or below some agreed datum level.
w = lb. per cub. ft. of water = 62·5.

Since the axis of the meter is horizontal, $h_1 = h_2$.

Hence (1) may be written

$$\frac{p_1-p_2}{w}=\frac{v_2^2-v_1^2}{2g} \qquad . \quad . \quad (2)$$

A difference in level of 15·5 in. of mercury is equivalent to

$$\frac{15\cdot5}{12}(13\cdot56-1) \text{ ft. of water} \qquad . \quad (3)$$

$$= 16\cdot223 \text{ ft.} = \frac{p_1-p_2}{w}$$

As we know the throat and full bore diameters we can state

$$\frac{v_1}{v_2}=\frac{d_2^2}{d_1^2}; \text{ or, } v_1=\frac{v_2\times d_2^2}{d_1^2}$$

$$\therefore\ v_1 = v_2 \times \frac{25\times4}{16\times49}$$

$$\text{or, } v_1^2 = 0\cdot01627 v_2^2$$

Substituting in (2) we may write

$$16{\cdot}233 = \frac{v_2^2 - 0{\cdot}01627 v_2^2}{64{\cdot}4}$$

$$\therefore \; v_2 = 32{\cdot}6 \text{ ft. per sec.}$$

Flow per minute through the meter

$$= \frac{d_2^2\pi}{4} \times v_2 \times 60 \times 6{\cdot}25 \text{ gal.}$$

$$= 14{,}970 \text{ gal.}$$

N.B. The reader may note that we have used the factor $(13{\cdot}56 - 1)$ in (3), and not simply $13{\cdot}56$. To show that this is correct, consider the two level sections a_1a_2 and b_1b_2, Fig. 151. The pressure at a_1 exceeds the pressure at a_2 by the difference between the weights of a column of mercury and a column of water, each of the same cross-section and height; this is equal to the weight of a column whose sp. gr. = (sp. gr. of mercury – sp. gr. of water) $= (13{\cdot}56 - 1)$.

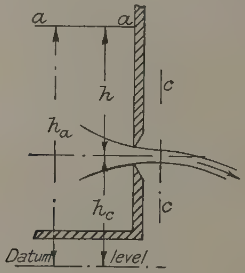


FIG. 152

FLOW OF WATER THROUGH AN ORIFICE

If water is contained in a tank and a sharp edged orifice be made at some distance h below the free surface of the water, then flow will take place somewhat as shown in Fig. 152. Observation of such a jet will show that there is a section cc which is considerably smaller than the orifice and, at this section, the stream lines are horizontal. It is this section cc, termed the *vena-contracta* which we have to consider in estimating the flow through the orifice. An application of Bernoulli's equation to the free surface aa and the *vena-contracta* cc enables us to write

$$\frac{p_a}{w} + h_a + \frac{v_a^2}{2g} = \frac{p_c}{w} + h_c + \frac{v_c^2}{2g} \quad . \quad (4)$$

Now $p_a = p_c$ = atmospheric pressure; $h_a - h_c = h$, and v_a is so small compared to v_c that it may be considered negligible; hence (4) may be written

$$h = \frac{v_c^2}{2g} \quad . \quad . \quad . \quad (5)$$

Expression (5) is the algebraic statement of *Torricelli's theorem*, which says that the theoretical velocity of flow through the *vena-contracta* is equal to that which a body would attain in falling in a vacuum through a height h, h being equal to the depth of the orifice below the free surface of the water.

By means of (5) we may calculate the velocity of flow through the *vena-contracta* section cc. This velocity, however, is never attained, since both viscosity and air resistance are encountered by the moving water, and so we find it necessary to introduce an *experimentally determined coefficient* k_1, termed the *coefficient of velocity*, and so we may write (5) as

$$v_c = k_1\sqrt{2gh} \quad . \quad . \quad (6)$$

We have already noted that the section at cc is less than the section at the orifice and, again, this requires the introduction of a second *experimentally determined coefficient* k_2, the *coefficient of contraction* which is numerically equal to the ratio

$$k_2 = \frac{\text{area of \textit{vena-contracta} section}}{\text{area of orifice section}}$$

The total flow Q per unit of time is given by

$$Q = A \times k_1 \times k_2 \times \sqrt{2gh}$$

or, $Q = A \times K \times \sqrt{2gh}$

A = area of orifice
$K = k_1 \times k_2$ = coefficient of discharge.

The following values of k_1 and k_2 are those usually accepted—

Type of orifice	k_1	k_2	K
Sharp edged circular	0·97	0·64	0·62
Projecting pipe	0·82	1·00	0·82
Re-entrant pipe (Borda's mouthpiece)	0·97	0·5	0·485

EXAMPLE. Water flows through an orifice under a head of 100 ft. If the coefficient of velocity = 0·97, find the height to which the jet would rise. If the diameter of the orifice = 1¼ in., and the coefficient of contraction = 0·64, determine the discharge in cub. ft. per minute.

Actual velocity $= v$

$$= 0{\cdot}97\sqrt{2g \times 100} = \sqrt{2gh_2}$$

or, $(0{\cdot}97)^2 \times 2g \times 100 = 2gh_2$

$\therefore \qquad h_2 = 0{\cdot}97 \times 0{\cdot}97 \times 100$

or, $\qquad h_2 = 94{\cdot}09$ ft.

$$Q = A \times k_1 \times k_2 \times \sqrt{2gh}$$

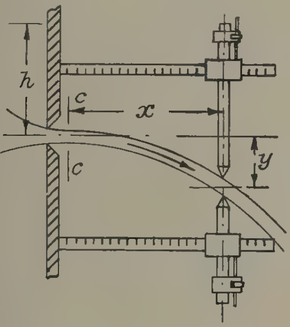


FIG. 153

$$Q = \left[\frac{25 \times \pi \times 0{\cdot}97 \times 0{\cdot}64}{16 \times 4 \times 144}\sqrt{2g \times 100}\right]$$

cub. ft. per sec.

or,

$Q = 0{\cdot}4246$ cub. ft. per sec.

$Q = 25{\cdot}476$ cub. ft. per min.

EXAMPLE. Oil, whose sp. gr. is 0·8, is discharged from a vessel in which the pressure is 60 lb. per sq. in. absolute to the atmosphere, pressure 14·7 lb. per sq. in. The orifice is 2 in. in diameter and the coefficients of velocity and contraction may be taken as 0·82 and 0·98. Calculate the velocity of discharge and the total discharge in tons per hour.

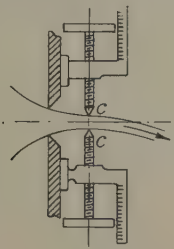


FIG. 154

$$\frac{p_1}{w} + h_1 + \frac{v_1^2}{2g} = \frac{p_2}{w} + h_2 + \frac{v_2^2}{2g} \quad . \quad (6)$$

Also, $h_1 = h_2$ and $v_1 = 0$; $\therefore$ we may write (6) as

$$\frac{p_1 - p_2}{w} = \frac{v_2^2}{2g}$$

$$\therefore \qquad v_2 = \sqrt{\frac{144\,(60 - 14{\cdot}7) \times 64{\cdot}4}{0{\cdot}8 \times 62{\cdot}5}}$$

$= 91{\cdot}66$ ft. per sec.

$Q = A \times k_1 \times k_2 \times v_2$ cub. ft. per sec.

or,

$$Q = \frac{\pi}{144} \times \frac{0{\cdot}82 \times 0{\cdot}98 \times 91{\cdot}66 \times 60 \times 60 \times 62{\cdot}5 \times 0{\cdot}8}{2240}$$

or,

$Q = 129{\cdot}12$ tons per hour.

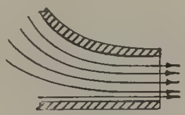


FIG. 155

DETERMINATION OF THE COEFFICIENTS k_1 AND k_2

To determine the coefficient of velocity, k_1, for any particular orifice, water is allowed to flow through the orifice under a constant head, and the path of the stream mapped by means of a depth gauge, as shown in Fig. 153. The path of the issuing jet is a parabola. If v_c is the velocity of the jet measured horizontally and assumed to remain constant, then

$$x = v_c \times t \quad . \quad . \quad . \quad (1)$$

Also, since the whole motion of the water at section cc is horizontal, its vertical fall after passing this section is given by

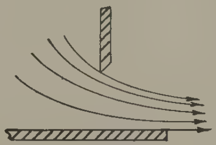


FIG. 156

$$y = \tfrac{1}{2}gt^2 \quad . \quad . \quad . \quad (2)$$

Combining (1) and (2) we obtain

$$y = \tfrac{1}{2}g\frac{x^2}{v_c^2}$$

$$\text{or} \quad v_c = x\sqrt{\frac{g}{2y}} \quad . \quad . \quad . \quad (3)$$

But, the theoretical velocity due to the head h is given by $v = \sqrt{2gh}$

$$\therefore\ k_1 = \frac{v_c}{v} = \frac{x\sqrt{\frac{g}{2y}}}{\sqrt{2gh}} = \sqrt{\frac{x^2}{4hy}}$$

The usual method of determining the shape and size of the stream at the *vena-contracta* is to arrange a number of screws with vernier scales, and to adjust these screws so that their points just touch the surface of the jet (Fig. 154). The area of the cross-section of the *vena-contracta* so obtained is used to determine

$$k_2 = \frac{\text{area of section } cc}{\text{area of orifice}}$$

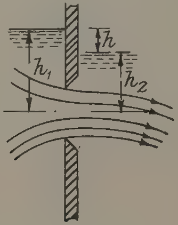


FIG. 157

Having determined k_1 and k_2, the product $k_1 \times k_2 = K$, the coefficient of discharge. This coefficient of discharge, however, is best determined by the direct measurement of the time required to fill a vessel of known and suitable capacity. This has been done by a number of reliable experimenters, and the reader is referred to the original papers and treatises for these values.

SUPPRESSED CONTRACTION. In order that all contraction of the issuing jet may be suppressed, the stream lines must be caused to flow in a direction normal to the orifice, that is, the stream lines become parallel. This can be brought about either wholly, as in Fig. 155, or partly, as in Fig. 156.

DROWNED ORIFICES. If an orifice is submerged on both sides, as in Fig. 157, it is termed a *drowned orifice.* The *flow* through such an orifice is then proportional to $\sqrt{h_1 - h_2}$

$$\text{or,}\quad Q = KA\sqrt{2g\,(h_1 - h_2)}.$$

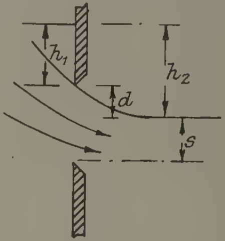


FIG. 158

When the orifice is only partly submerged, as in Fig. 158, the total flow may be taken as the sum of (1) the flow through the unsubmerged section of depth d, and (2) the flow through the submerged section of depth s, and can be stated as $Q = Q_d + Q_s$

$$= K_d A_d \sqrt{2g\,\frac{(h_2 + h_1)}{2}} + K_s A_s \sqrt{2gh_2}.$$

POWER TRANSMISSION BY CHAINS

By H. M. Andrew, Wh.Ex, A.M.Inst.C.E.

LESSON II

TYPES OF DRIVING CHAIN

Of the four principal types of driving chain in common use at the present time, namely, malleable cast link chain, block chain, bush roller chain, and inverted tooth chain, the first two are intended principally for conveying purposes, while the last two are designed solely for power transmission.

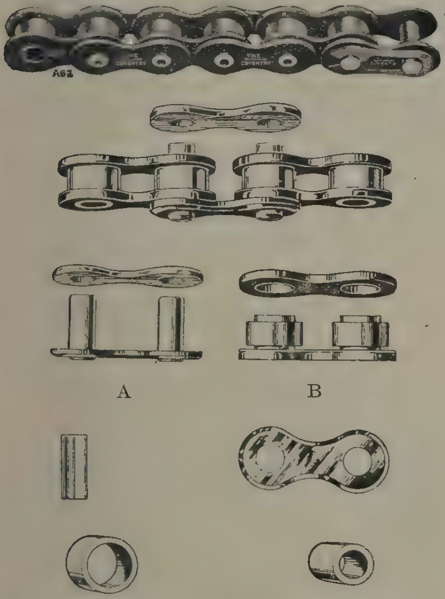


Fig. 1. Bush Roller Chain and its Component Parts

Malleable Cast Link Chain has considerable use in some industries for slow speed low power drives, because of its cheap construction. In a modified form, with attachments cast on to the links, and sometimes with pin bearings, it is extensively used for conveying purposes, but its weight to strength ratio is sufficiently high to restrict its use considerably for power transmission.

Block Chain has alternate pitches formed as solid blocks of metal to facilitate the attachment of conveying members. This, of course, renders this chain also too heavy for a given strength when used purely for transmission of power. Only the alternate pitches engage the wheel teeth and consequently its gearing is not so smooth as the bush roller type, which has largely superseded it for power drives. Its bearings, unlike the malleable cast chain, are fully machined and securely enclosed, but the solid metal block does not lend itself easily to the formation of a properly hardened bearing for high speed driving purposes (the Coventry Chain Co. overcome this by laminating the block).

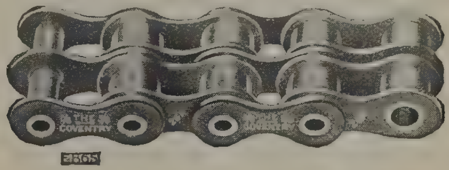


Fig. 2. Duplex Bush Roller Chain

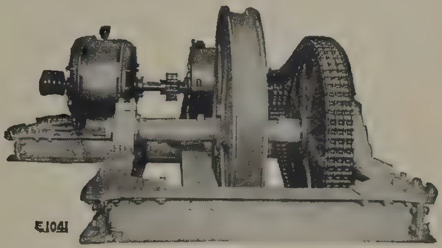


Fig. 3. 60 H.P. Drive using Coventry Triplex Bush Roller Chain

Almost all racing cyclists formerly used this type of chain for pedal cycle track racing, but nowadays bush roller chain is largely employed.

Bush Roller Chain. Chain makers generally manufacture this well-known type of driving chain on somewhat similar lines, differing mainly in material and small details of assembly. The principal dimensions have been standardized in England and also in the U.S.A., but these standards have not yet been universally adopted.

This chain consists of cylindrical bearing pins (Fig. 1) riveted to flat side links (generally of figure eight shape), forming one link or "pitch" of the chain, *A*, the intermediate links or pitches, *B*, comprising tubular bushes surrounding, and forming a bearing for, the above bearing pins; these bushes being forced into similar, but larger, side links. Surrounding the bushes, and free to rotate on same, are the rollers. Fig. 1 shows all these parts in detail.

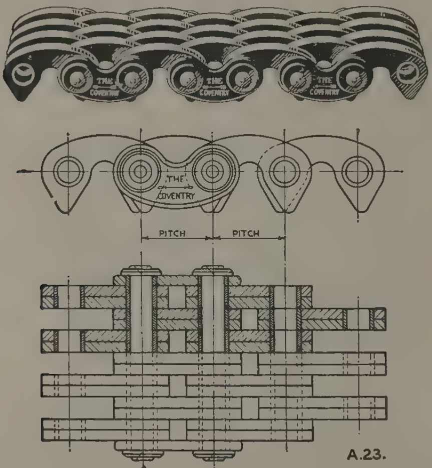


FIG. 4. CIRCULAR BUSH INVERTED TOOTH CHAIN

The shear strength of the bearing pins should, in a balanced design, be equal to the tensile strength of the side links, but is often rather greater in order to give an ample bearing surface. Owing to the need of a working clearance, the bearing surface commences theoretically as a line contact of the pin across the full width of the bush, until wear in service has developed a full arc of bearing surface. The closer the initial fit of the pin in the bush, the sooner the full arc of bearing surface is established, an advantage gained by accurate workmanship. This type of driving chain gives—

(*a*) Links directly in line of pull.

(*b*) Fixed bearing pins supported only at the ends.

(*c*) Bearing surface across full width of inside link, line contact at first but gradually wearing to a full bearing after about 0·6 per cent extension by wear has taken place.

(*d*) Wear in the bearings tends to lengthen the alternate pitches only because the "pitch" of the chain, for gearing purposes, is the distance from face to face of adjacent rollers, and the pairs of rollers mounted on pitch *B* do not, of course, alter their centre distance by wear.

(*e*) The amount of bearing surface in any particular pitch of chain is governed by the available diameter for the bearing pin and the maximum practicable length that such a pin can have when supported only at the ends.

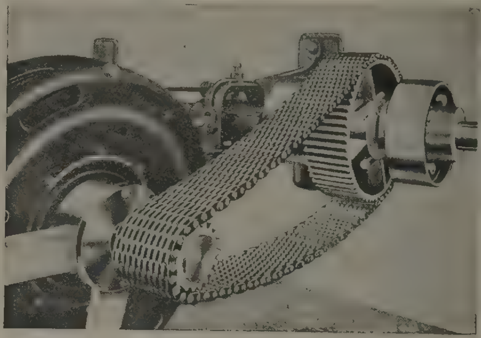


FIG. 5. LINESHAFT DRIVE BY INVERTED TOOTH CHAIN

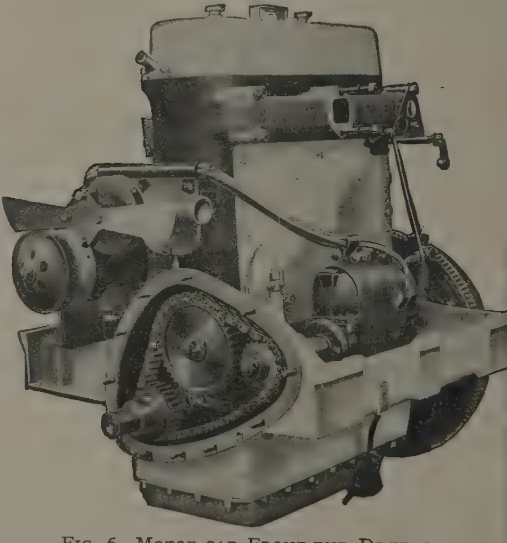


FIG. 6. MOTOR-CAR FRONT-END DRIVE BY INVERTED TOOTH DRIVE

NOTE. The bush roller type of chain is, however, manufactured in duplex form (Fig. 2) to obtain increased width for a given pitch, the centre links being made as tight a fit as possible on the bearing pins, so as to give central support. Some makers even manufacture it in triplex form, as shown in the 60 h.p. Coventry chain drive illustrated in Fig. 3.

Inverted Tooth Chain. This type of chain is made in a variety of designs by different makers, the variation being mainly in the construction of the bearing, but also to some extent in the shape of the links.

In its simple form (Fig. 4) it has a cylindrical bearing pin supported by links at several points along its length. These links are of arched form so as to pass over the wheel teeth, which make contact with and drive against the end faces of the links. The links of adjoining pitches are interleaved with one another so that half the links on any one bearing pin belong to one pitch of the chain, and the other half to the next pitch; these two sets of links being in tension and pulling against each other. The working bearing surface is therefore that due to one set of links only, that is, it extends over only half the length of the bearing pin. The *diameter* available for the bearing pin is limited by the necessity of restricting the size of the link heads to the space between the wheel teeth, but the *length* of this pin can be the longest practicable hardened pin for the given diameter.

Fig. 7. Motor-cycle Drive
(Coventry "Ultimate" Chain)

This type of driving chain gives—

(*a*) Arched links with the crown of the arch just above the line of pull.

(*b*) Bearing pins that are free to rotate and which are supported by several links at regular intervals along their length.

(*c*) A much wider chain available for any given pitch than in the case of single strand bush roller chain, but with bearing surface across half its width only. The pin diameter is necessarily rather smaller than can be utilized in the bush roller type.

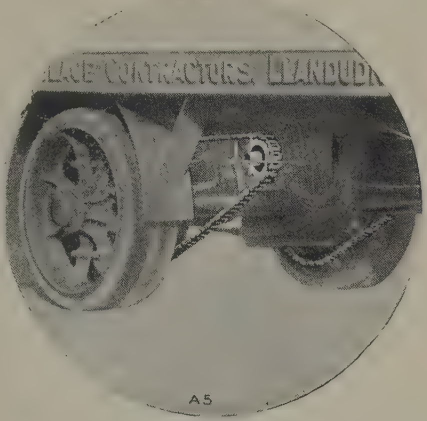


Fig. 8. Rear Drive on Commercial Vehicle

(*d*) Wear causes all the pitches to lengthen equally.

(*e*) The method of gearing with the wheel teeth tends to minimize the sound produced by contact between chain and wheel, as explained in Lesson I.

Bush roller and inverted tooth chains are so different in their characteristics that each type has so far managed to hold its own as a driving medium. Inverted tooth offers a quieter and smoother drive than bush roller at ordinary industrial speeds and, in spite of its higher cost of production, is largely used for lineshaft drives (Fig. 5), motor-car camshaft drives (Fig. 6) and similar applications where a quiet drive is essential, and first cost is not the governing consideration. The more robust design of the bush roller chain, with its superior capacity to resist shock and fatigue, has standardized this type for motor-cycle (Fig. 7) and pedal-cycle drives, rear drive on commercial vehicles (Fig. 8), and for heavy duty in exposed situations, such as hot saw drives, rolling mill drives, concrete mixers, etc. Bush roller chain is, however, reasonably quiet under certain suitable conditions and, this fact, together with its relative cheapness, has led to its occasional use recently for motor-car cam-shaft drives and lineshaft drives formerly considered to be the exclusive field of the inverted tooth chain.

While a detailed description of each of the existing types of inverted tooth chain would be out of place in these articles, it will be as well to give here a brief note on the principal designs.

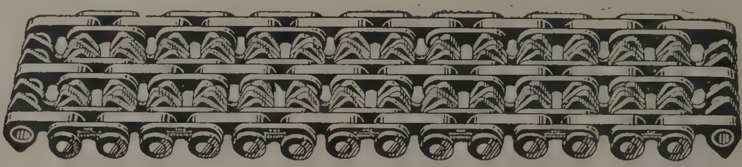


FIG. 9. COVENTRY "BIFLEX" CHAIN

The Circular Bush type illustrated in Fig. 4, has short hardened bushes forced into double links to form the bearing. The angle between the two end gearing faces of any one link is customarily 60°, although both 60° and 75° are made by the Coventry Chain Co. The larger this angle, the larger can be the bearing pin. The smaller this angle, the quieter the chain in operation. The links are made with curved back, as shown in Fig. 4, or with flat back, a shape illustrated in the "Biflex" chain (Fig. 9), in which type of chain half the gearing links face upward and half downward so as to enable this particular chain to gear with chain wheels on either face of the chain, as in the case of bush roller chain.

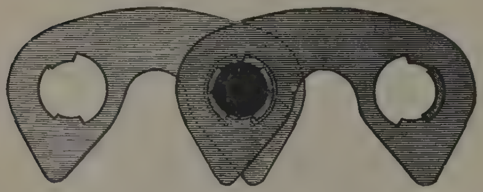


FIG. 10. LONG LINER INVERTED TOOTH CHAIN

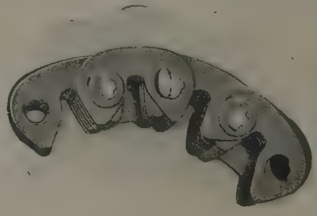


FIG. 11. MORSE CHAIN

The Liner type of inverted tooth chain is an early design that aimed at securing bearing surface across the full width of the chain by using a long segmental liner that is common to all the links in one pitch. As shown in Fig. 10, a pair of these liners fit either side of each bearing pin, one liner being a force fit in one pitch of links and the other a force fit in the adjoining pitch of links. The holes in each link head, in addition to receiving the bearing pin, allow space for one liner as a force fit in the link head and the second liner with clearance round it to permit of its rocking around the bearing pin. As the space between the wheel teeth for the link head is a fixed quantity, the bearing pin in this design must be slightly smaller in diameter than in the circular bush design, and the arc of the bush is limited to about 110°.

The Segmental Bush inverted tooth chain is a design produced by Messrs. Renold, Ltd., that uses a short segmental liner (in place of the circular bush), that is forced into a pair of links. This enables a larger diameter of bearing pin to be employed for a given size of hole in the link head. Like the circular bush type it has bearing surface across only half the width of the chain, but an attempt is made to make the segmental bush as exact a fit as possible on the pin so as to secure maximum initial bearing surface. The arc of the bush is 140°.

The Plain Bearing type uses no bush at all in the link heads, which are punched to fit the pin, the links being locally hardened or toughened as far as they can be without appreciably affecting their tensile strength or fatigue resistance.

The Link Belt Co. (U.S.A.) in a recent design secure more than half bearing surface across the width of the chain by making the alternate pitches of narrow extra strong links formed as a tight fit on the bearing pin, the intermediate pitches being comparatively broad links interleaved with the above and having a wide bearing surface on the pin. This advantage has the accompanying disadvantage that the pin cannot rotate and, therefore, the wear is concentrated on one arc of the pin.

The Whitney Chain Co. (U.S.A.) design aims at a *double* bearing. The alternate pitches consist of two inverted tooth links forced on to the ends of two circular bushes. The intermediate pitches are formed by two plain bearing inverted tooth links having their bearing on the surface of the above bushes between the force fit links of the alternate pitch. Unit chains of this build are connected in multiples of the unit width by passing bearing pins through the bushes, and coupling these bearing pins by means of fixed side guide links (in the same pitch as the intermediate plain bearing links), thereby forming a second inner bearing across the full width of the chain.

The Morse Chain Co. (U.S.A.) and the Westinghouse-Morse Chain Co., Ltd., of England, manufacture the Rocker joint type of inverted tooth chain exclusively, in which a rocker attached to the links in one pitch (Fig. 11) rocks or rolls on a seat pin attached to the links of the adjacent pitch. In the straight length of chain between chain wheels, the flat side of the rocker is designed to lie on the flat side of the seat pin, with a view to preventing relative movement while passing from wheel to wheel. Various link angles have been employed in Morse designs. In some designs two different angles are employed at the two ends of each individual link. Gearing with the waist of the link as well as the end faces is also a feature of some of these designs; the object aimed at being to minimize noise.

Fig. 12 illustrates multiple strands of this chain in a high powered hydro-electric drive.

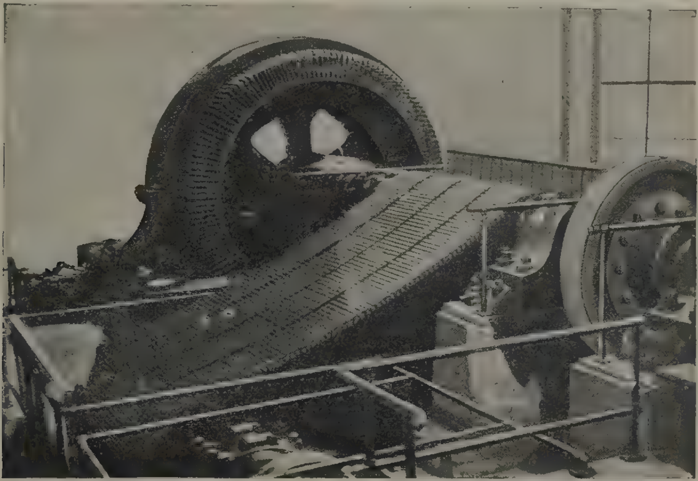


Fig. 12. Morse Chain in High-power Drive

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XXVI

Visualizing. The guidance drawing as a means of reference in preparing the design of a machine part has a useful ally in the designer's memory, and the alert engineering student will be advised to take every opportunity to inspect various designs of machinery. Such observation of existing design stores the mind with useful ideas of detail design together with a sense of proportion.

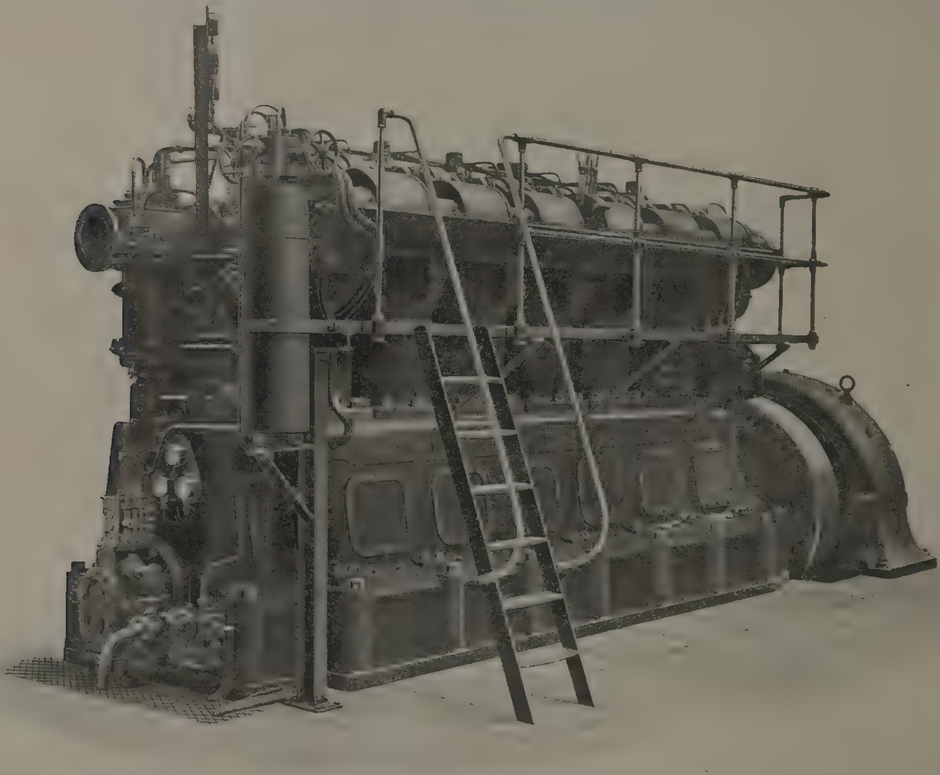


Fig. 2. 600 B.H.P. Four-stroke Cycle Diesel Engine
(Messrs. Belliss and Morcom, Ltd.)

The above sense of proportion is sometimes developed to a very fine degree, enabling an experienced designer often to pass judgment upon the strength of a machine part independent of calculation. Such judgment is, of course, only reliable within certain conditions, e.g. where the strength of a shaft is in question, opinion regarding its power carrying capacity will only hold providing the speed of rotation lies within fairly narrow limits.

To exploit the engineering experience of one's memory it is necessary to acquire the art of visualizing the machine part represented on the drawing. With practice it is possible to read the three views of a drawing and visualize a

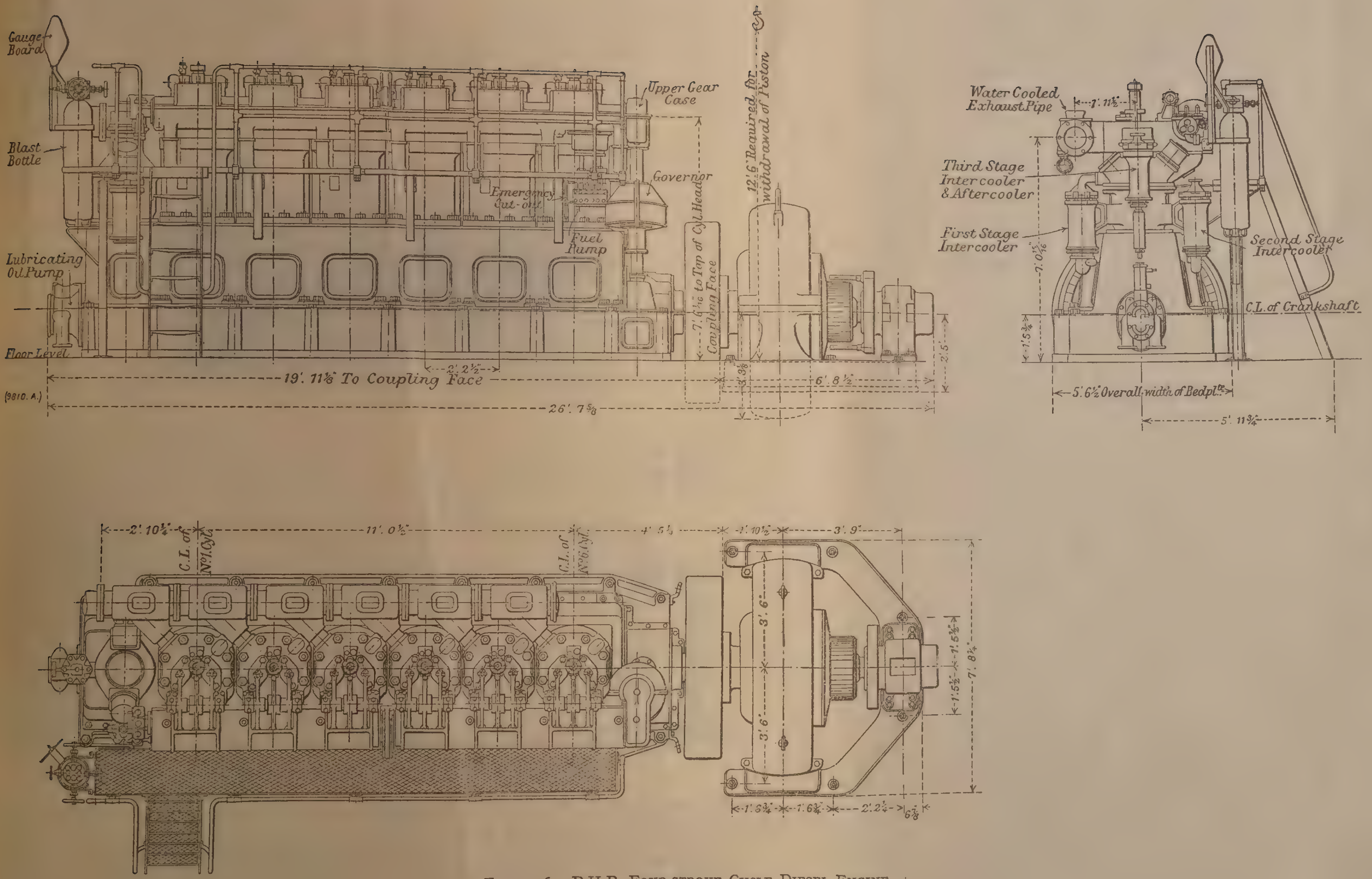


Fig. 1. 600 B.H.P. Four-stroke Cycle Diesel Engine
(Messrs. Belliss & Morcom, Ltd.)

perspective view of the part, and although the art comes with the practice of drawing it is advisable to cultivate it to a finer degree. This will, of course, follow if the habit is formed of visualizing the outline of a machine part both before and after making the drawing of the part.

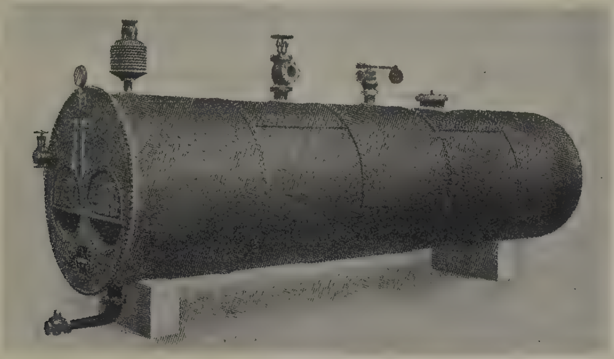


Fig. 3. Lancashire Boiler

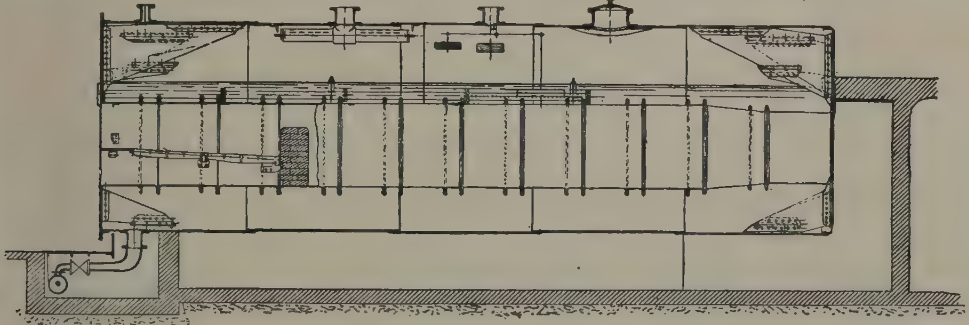


Fig. 4. Section of Lancashire Boiler

Visualizing to Scale. As an extension of the above we may acquire the sense of visualizing the form of a machine part to scale, i.e. mentally increasing or reducing the proportion according to the drawing scale in order to obtain a comparative idea of the size of the detail.

This is particularly necessary where we have to consider the space required for operating the machine or accessibility in terms of an averaged sized operator. Further, it is often necessary to have an idea of both the weight and size of the part when such has to be man-handled.

Engine Arrangement. An example of the above art is provided in Figs. 1 and 2, which show an arrangement of an engine together with a perspective view of the engine. The illustrations are taken from *Engineering* for 19th November, 1926, and refer to a 600 b.h.p., four-stroke cycle Diesel engine constructed by Messrs. Belliss & Morcom, Ltd.

Referring to Fig. 1, it will be seen that certain features of the design must be considered with reference to the size and convenience of the engineer operating the machine. For example, if before arranging the ladder and platform, the pressure gauges or the various controls, we visualize the engine as shown in Fig. 2, we may more readily decide the best position of these parts relative to the engine.

Boiler Arrangement. A further example is provided in the drawing of the Lancashire boiler (constructed by Messrs. Yates & Thom, Ltd.) shown in Fig. 3.

First, in arranging the level of the floor in front of the boiler it is necessary to estimate a suitable height of fire-bar for ease in stoking.

It is also necessary to observe the position of the floor in fixing the height of valves which may be operated from the floor level. Further, consideration must be given to the question of access to the inside of the boiler. The practice is, of course, to provide a manhole in a position free from obstruction both immediately outside and inside the boiler.

Referring to Fig. 3 and the sectional drawing of the boiler shown in Fig. 4, it will be observed that one manhole is arranged on the boiler front below the two furnaces. To check the value of this provision of access we must imagine both the space under the furnaces and the space occupied by a man moving along the boiler. A dimension of the tightest passages combined with a visualizing of the finished boiler will enable the draughtsman to decide upon the practicability of the design in this respect.

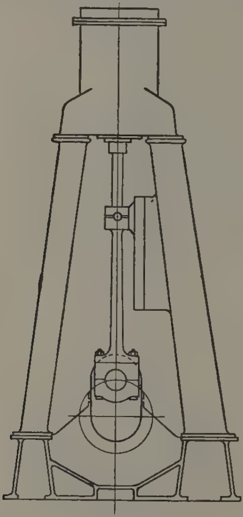


FIG. 5

Proceeding with the imaginary movement of the engineer along the boiler, we may consider the question of access to that part of the boiler above the furnaces. With one manhole only access to the upper part of the boiler would be possible only by means of the passage between the furnaces, which is a maximum at the rear end of the boiler, a second manhole is, therefore, provided in the steam space.

Visualizing Movement. Where the machine drawing represents moving parts it is necessary to observe that the movement can take place unchecked. Frequently, the clearance between the moving and fixed parts is sufficiently large to show at a glance. In other cases the clearance is too small to allow of so approximate a check as is provided by a look at the drawing, and it becomes necessary to make a geometrical check.

An example of a moving machine part passing very close to the fixed part is given in Fig. 5, which shows the bottom end of a reciprocating engine connecting rod and the adjacent columns and bed-plate.

It will be evident from the above figure that the inclination of the connecting rod will vary during the revolution of the crankshaft, and to check the bottom end of the rod for clearance throughout the movement it would be necessary to draw the outline of the moving part in its true position at many points in the revolution. Such a procedure may prove both lengthy and tedious, and much of the task may be dispensed with if we first determine that part of the cycle which provides clearance for any inclination of rod. Referring to Fig. 6 it will be seen that the rod, in moving from the vertical as the crank rotates, swings the lower end of the bearing closer to columns and bed-plate. Taking a radius r_1, sufficiently large to clear this end, and describing an arc cutting the rod centre line at a, then a circle of radius oa will provide a space to allow the rod to swing unchecked at any angle during the revolution. We observe this circle to lie clear of the adjacent fixed parts, and so we may conclude that the lower end of the rod is correct for clearance. Passing to a consideration of the upper end of the rod bearing, an arc of radius r_2 clearing this end and

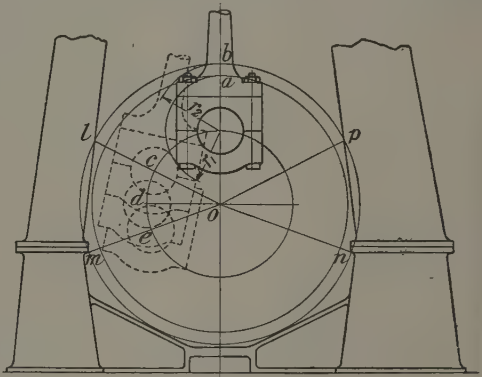


FIG. 6

cutting the rod centre line at *b* will provide a radius *ob* producing a circle in which the rods could swing on any angle during the movement unchecked within the circle. This circle, it will be seen, intersects the columns at *lm* and *np* indicating a zone of possible interference.

Before checking this part of the movement we may observe that the inside outline of the columns are symmetrical about the engine centre line, thus allowing us to pay attention to one side only. We have now limited our search to the angle *lom*. Proceeding, an outline of the rod drawn in its true position with the crank at the points *c*, *d*, and *e* will show that the rod will clear the fixed parts throughout the revolution of the crank.

WORKS ORGANIZATION AND MANAGEMENT

By E. W. Workman, B.Sc., A.M.I.E.E., etc.

LESSON XIII

THE EMPLOYMENT DEPARTMENT

Staff Requirements. In dealing with nearly every department mentioned in these lessons, it has been necessary to point out the importance of having on the staff men and women who have particular characteristics suited to the department to which they are attached. It is of the utmost importance to the welfare of a factory that the employees shall be *chosen* in the most efficient manner. The problem is always to get the right man in the right place, and if this is accomplished successfully there is a very much greater chance that the remaining problems of the factory will also be solved in a satisfactory and efficient manner. In fact, important as are the problems of good organization and good plant, the choosing of a good staff is probably of still higher importance. This is especially so, as equipment and methods of organization are becoming more and more standardized and are available to all competing firms alike, so that ultimate success will rest with the firms that choose and retain the best and most suitable staff.

Old and New Methods. As regards the old methods, it would be more correct to say that they did not exist. The engaging and dismissing of employees was a matter entirely in the hands of the foreman or head of the department, who might or might not have the gift of being able to choose his staff correctly without an undue amount of trial and error. The necessity of adopting better methods is apparent when one considers that every person is far more suited to a particular class of work than to any other that he may attempt. The difficulty is, of course, that a man very often is not aware of the type of work most suited to him as he has never had any opportunity of change from the work into which he originally drifted on leaving school. At the same time, if a man is not doing the work for which he is most suited, he is not working at his maximum efficiency, and both the man and the factory are losers financially.

Labour Turnover. This phrase is used to denote the amount of change in the personnel of a firm, due to the engagement and discharge of the individual employees. Why the subject has received so much attention of late is that it has been found to be a most expensive process, costing far more than is generally realized. Mr. M. W. Alexander, of the General Electric Company, carried out investigations in America, in 1912, in a large group of factories that employed 38,668 workers at the beginning of 1912, and 46,796 at the end of the same year. To attain this increase of 8,128 workers, 44,365 had been engaged during the year, i.e. about five and a half times the number of persons required. In investigating a labour turnover of 22,225 employees he found that the average cost to the firm was about £7 for each person engaged, for which he gave the following details—

1. Clerical work of hiring	2s.
2. Instruction given new employee	8s. to £4
3. Increased wear and tear, and damage of machinery and tools	4s. to £2
4. Limited output	£1 to £4
5. Spoiled work	£3

Other estimates ranged from £6 to as high as £30 or £40.

Reasons for High Turnover. The rate of labour turnover should be watched in every department, and if the rate appears excessive in any particular department, investigations should be made to try and discover the cause. The main reasons of a high turnover are as follows—

1. Wrong type of workers being engaged. If this is the cause of the high turnover, something is wrong with the organization of the employment department.
2. A foreman or head of a department with whom it is difficult to get on. Such officials are often very skilled technically, and it is often a difficult matter to determine whether the high degree of skill compensates for the difficulties that arise in connection with the department.
3. Bad working conditions. In this case it will usually be found cheaper to alter the bad conditions, rather than to incur the perpetual expense of a high labour turnover.
4. Outside causes. These may be subdivided into—
   (*a*) The offer of a better position.
   (*b*) Retirement because of old age.
   (*c*) Domestic reasons necessitating leaving the district.

The Employment Department. The offices allocated to the employment department must contain sufficient waiting room for applicants for work, and also a private interviewing room in addition to the clerical office where the employment records are kept. The department should be in charge of a responsible "Employment Manager." The work of the department consists of—

1. Supplying employees in answer to departmental demands.
2. Keeping down the labour turnover.
3. Keeping records of the engagement, promotion, and particular characteristics of each employee.

This work is better concentrated in one department because—

1. An applicant is more likely to be sent to the most suitable department.
2. Interviewing is better done by a specialist with an accumulated experience of engaging employees.
3. Appointments are more likely to be made for the general good of the firm, and with less private favouritism for friends.

Engaging Employees. The usual procedure is for the employment manager to interview every applicant for work and to make a record of the necessary particulars, such as name, age, trade, wages expected, etc. These particulars are recorded on a standard card, and sent with the man to the head of the department where it is proposed to place him. If engaged by this department, the card is returned to the employment department and forms the basis of the future record of the man's progress whilst in the factory. The employment department is responsible for notifying the wages department of the terms of the engagement. Similarly, when a man leaves or is dismissed, the notification is received by the wages department from the shops via the employment department. Another duty of the department is to see that all legal requirements are carried out in regard to the engagement—especially as regards women and children.

When the employment manager finds a good man who is not required at the moment, or who is not immediately able to accept an offer of employment, he should keep in touch with the particulars by means of special filing of the record card.

Follow-up Work. In addition to the particulars already mentioned, future transfers between departments should be notified to the employment department, so that the history record may be complete. By a study of these cards, the employment manager may learn a great deal that will guide him in the future engagement of employees, and which will show him wherein his judgment tends to err.

Apprenticeship. Some organization is needed to deal with questions of apprenticeship, and this is very often undertaken by the employment department as the work is closely allied to its own. On the other hand, a separate education department may exist, in which case this is the right place to deal with apprenticeship. In some works the education department comes under the same head as the research department, this arrangement being of considerable value in the case of the more highly-educated apprentices who can do advanced research work.

Apprentices may be divided into two categories—

1. Trade apprentices, whose object is to become skilled tradesmen.

2. Technically educated apprentices, whose object is to gain a practical training before being appointed to one of the higher staff positions. Some firms have well-organized schemes in operation for this class of apprentice, whereby they are engaged immediately after obtaining a technical degree. In the course of a two or three year training course such students are given experience in a variety of departments, and at the end, if suitable, they are made the offer of a staff position. This procedure ensures that such a firm has its responsible posts filled by men with good education, and a broad knowledge of the ramifications of the departmental organization of the particular firm.

Lectures. For both types of apprentices, suitable courses of lectures may be arranged, and these may be given either during ordinary works hours or in the evenings. Such courses of lectures may be supplemented by lectures given by heads of departments immediately after the works normally close, and available to any employee who cares to attend. These lectures would deal with the work of the department to which the lecturer was attached, and questions would usually be replied to at the close. A series of lectures on these lines not only serves a valuable educational purpose but, at the same time, tends to make smoother the relationship of employer and employee.

Conclusion. In these last eight lessons the organization and duties of a number of departments have been discussed. The departments selected probably represent the most important that will be found in any engineering works, but it must be emphasized again that whether more or less departments are found in any particular firm depends on the particular local conditions that exist and determine the most suitable organization. Works organization can never be stereotyped, if it is, it becones inefficient and dead, and ceases to be worthy of the name. Routine must be established, but the routine must always be open to change and improvement when new conditions demand, and the most suitable routine will often vary somewhat according to the people who deal with it. Thus, works organization and management is a study, a practical study, and a study that can never be ended. Its object is to make the best of men and material, and in doing this, the very best also will be demanded from the one who organizes and manages.

(CONCLUSION)

MACHINE TOOLS

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XX

TURRET LATHES

Definitions. The term "turret," following that of the "monitor," originated in America, named after the gun turrets in battleships. It embraces all those designs in which a battery of tools is carried in a revolving body. Over here, a distinction is made between the turret and the English term "capstan" lathes. A capstan slides along a base which is set on and clamped to the lathe bed; a turret fits on a saddle that slides along the bed, carrying the turret with it. The latter has advantages which secure its popularity over the other, because the length of travel of a capstan is limited by that of its base, while that of a turret may be as much as the length of bed will permit. A long travel is very desirable when long boring bars with pilot ends have to be drawn back far enough to clear the cross slide, and to be rotated to bring succeeding tools into action. A typical turret lathe is shown by Fig. 135 by the Warner & Swasey Company, of Cleveland, Ohio, U.S.A. Similar machines are made with all-geared heads. The utilities of the common lathe are frequently extended by mounting a capstan on the carriage of the slide rest to carry from four to six tools. A simple locking device for each station is operated with a hand lever. There are many diversities in these forms.

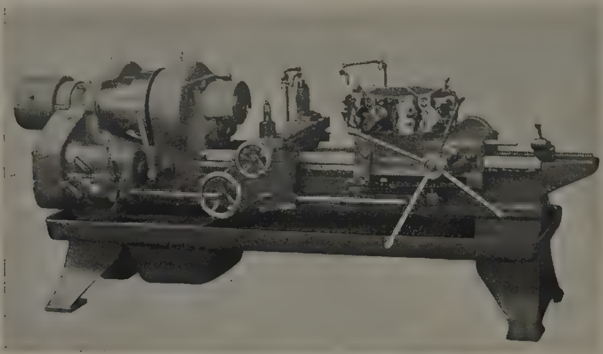


Fig. 135. Hexagon Turret Lathe

Brass Finishers' Lathes. These are still generally hand-operated, and they commonly include a chasing bar pivoted at the rear for cutting short screw threads, the lead being derived from master hobs slipped over an extension of the mandrel. The mandrel is hollow to receive the stock bars thrust in from behind, and having the loose end supported on a floor stand. In most cases a capstan has axial movement, only, but a cross slide is frequently included in brass finishers' lathes for facing operations. This may be lever-operated only, or a slow feed may be included with a screw. Stops control the limits of longitudinal and cross feeds.

The Capstan, or Turret. The fitting of this element to its base is done in various ways, the main object being to avoid slackness due to wear, to provide secure locking, and stability with heavy overhanging tools. Most heads have the holes that receive the tool shanks pierced right through to permit of the passage of stock bars through when the tool is mounted to one side. For this the capstan must have an even number of holes. A few capstans are tilted so that the longer tools clear the top of the cross slide without a large range of backward movement.

A simple form of hand-operated capstan is shown in the group of Fig. 136, but designs are modified in many different ways. In this example the capstan *A* is circular, pierced with six holes. A lever, fulcrumed in the pivoted boss *C* and passing and sliding through the swivel boss *B* moves the capstan slide along its base. A ratchet *D* imparts one-sixth of a turn to the capstan on the backward stroke, when the pivoted lever *E* makes contact with one of the teeth. Then it is locked with a spring plunger *F* that enters one of the tapered slots in the base of the turret. The unlatching of the capstan is effected by the pivoted dog *G*, one end of which enters a slot in the plunger *F*, the other end carrying a roller that comes in contact with a stop plate *H*, screwed to the base. The upper end then draws the plunger back in opposition to the spring at its tail end. Conversely, when the slide is fed forward to bring the tool up to its cut, the ratchet is disengaged from the lever *E*, the roller leaves contact with the stop plate, and the spring at the end of *F* forces it along until its tapered end enters and locks in a notch in the indexing plate. A tapered gib prevents risk of the development of lateral play in *F* with inaccurate indexing. The turret pin is tapered to take up wear. Parts in frictional contact are hardened. Details of locking arrangements differ much in various designs, and the method of gibbing the turret slide to its base. The circular capstan is used only in the smaller lathes. Others have flat facets, which are desirable, because they afford abutments to which heavy tools and boxes of tools can be bolted firmly, a security which is lacking when a stem is only clamped in a round hole. Always the work of these lathes tends to become heavier, and the combinations of tools more intricate, and heavier.

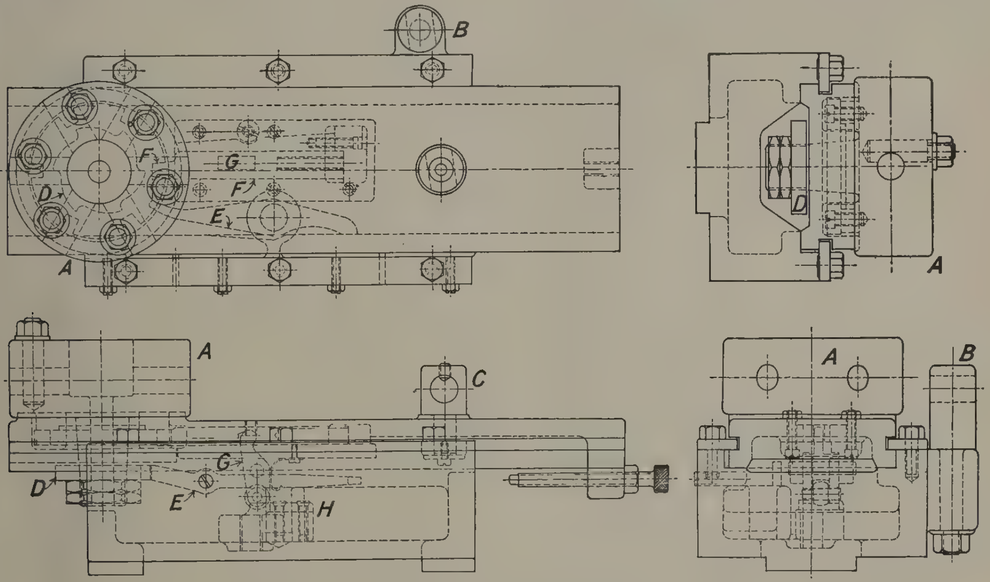


Fig. 136. Mechanism of Hand-operated Capstan

In Fig. 136 a single adjustable stop is seen, which locates the termination of a cut without tentative measurements. But to use one stop, to arrest the cut of, say, half a dozen separate tools means that the length of travel of each tool must be accommodated to the limit of travel of the stop. This entails much tedious setting of every tool when shoulders must occupy accurate relations. A design therefore now more commonly adopted is that of mounting a separate stop for each tool. The stop bars, screwed for adjustments, are carried in a revolving disc at the rear of the slide, and each is set to suit the travel of its tool the movement of which it arrests. Automatic mechanism operated by the revolution of the capstan causes a partial rotation of the disc to bring the stop into contact with an abutment face on the base. Hexagonal bars with slots with stop dogs are often used in front of the bed. In the turret lathes proper, where the turret and its carriage

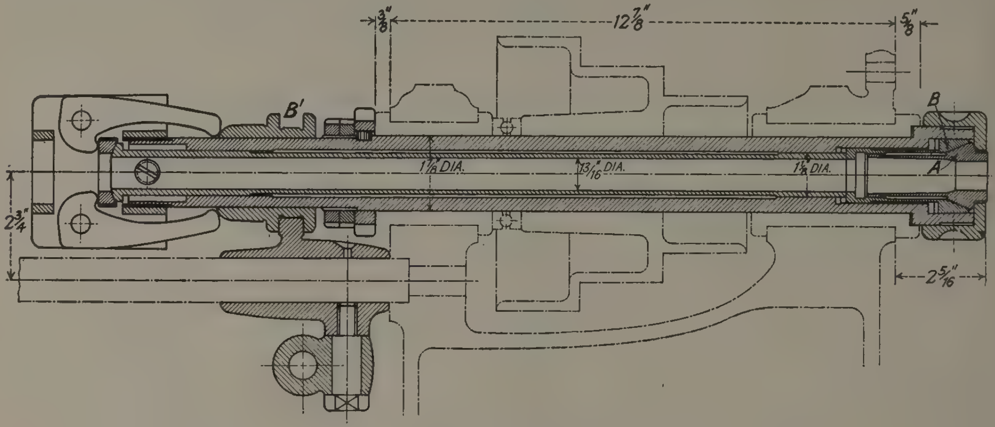


FIG. 137. PUSH-OUT COLLET MECHANISM

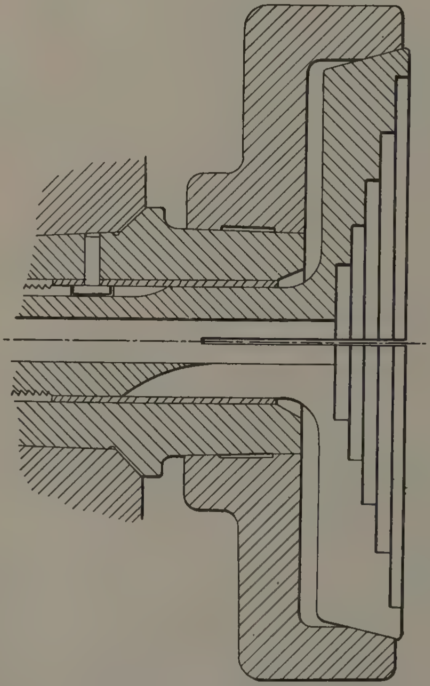


FIG. 138. SPLIT COLLET FOR DISCS

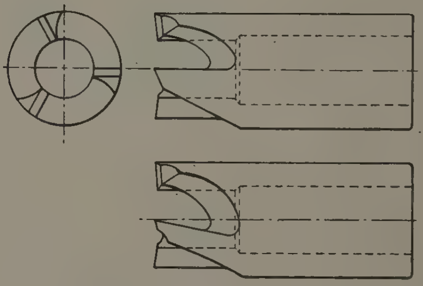


FIG. 139

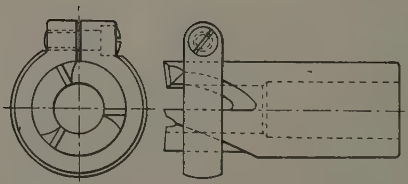


FIG. 140. HOLLOW MILLS: FOR BRASS, FOR STEEL, AND ADJUSTABLE TYPE

slide together along the bed, the longitudinal motion is derived from a splined feed rod actuated from the headstock. When the movement is arrested by a stop rod, the effect is to throw a drop worm out of action, and so trip the feed and break the connection between the feed rod and the slide.

Headstocks. The headstock for a capstan or turret lathe has a hollow spindle for the passage of bars from the back, introduced on the early screw machines, with the Parkhurst wire feed, previous to which single pieces could only be chucked from the front. It also includes some form of collet chuck for the gripping of bars after they have been fed through a predetermined distance from the rear. Some heads are designed for dealing with both bar and chucked work, but the tendency now is in favour of limiting the function of a machine to one or the other.

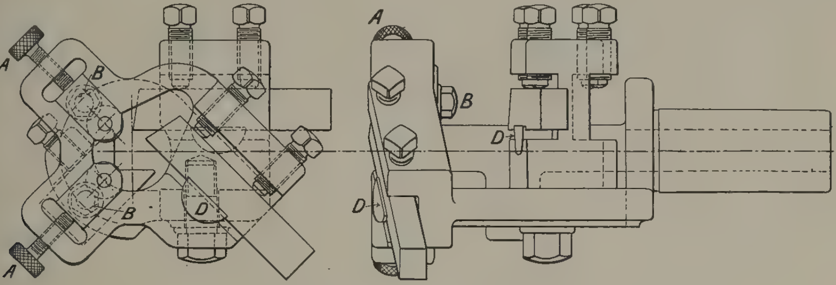


Fig. 141. Double Cutter Box-tool

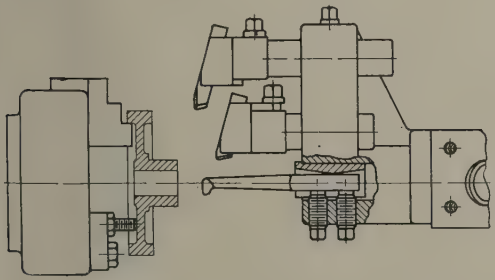


Fig. 142. Simultaneous Turning and Boring

Fig. 137 is a typical spindle with collet *A* closed with a tube of the push out type, in some designs the tube is drawn back. It is pushed along to the right for gripping, and moved to the left for release. The collet is coerced concentrically by the coned ends of the encircling cap *B* which is attached to the draw tube. The sliding is produced by a pair of lever dogs at the extreme left, fulcrumed in a collar on the tail of the spindle, and put into action by the thrust of the coned collar *B′* against the long arms of the dogs, when the short arms press against the end of the draw tube, thrusting it to the right and closing the collet on the bar. Moved in the opposite direction the pressure is released, and the elasticity of the collet permits it to open sufficiently to permit the bar to be fed. The coned collar is slid with a lever in the bracket below, moving on a rod and engaging the slipper in the annular groove in the cone. This is moved with a hand lever, or automatically with a suspended weight, by power in a large group of machines. The stock that passes through the hollow spindle is supported at the left-hand end in a bracket on a continuation of the bar that carries the feeding bracket, or on a floor stand at the rear, when feeding is effected with a lever, or with a suspended weight. The capacity for bar work is the size of the hole in the hollow draw tube, but articles put in from the front can be much larger by employing adapter chucks, of which there are many designs. An extreme example is shown by Fig. 138, for gripping large discs.

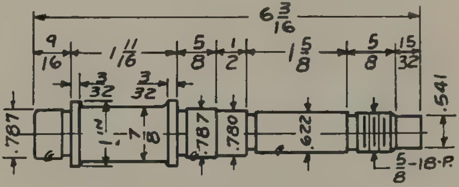


Fig. 143. Fan Shaft

The Cross Slide. This, originally a cut-off rest, now fulfils many functions, among which the severance of pieces finished from bar is of lesser importance. An example is seen in Fig. 135. Tool posts are carried at front and back on a top slide that has a transverse movement across the carriage, which may be clamped on the bed, or be traversed for turning or screw cutting. The tool posts are fed inwards for cutting off, for forming irregular outlines, for surfacing and knurling. Cross movements are effected by hand or automatically. The front and back tools holders are operated independently or in unison, with a hand lever or a screw. Stops are fitted to arrest the movements of the tool posts for sizing. Tool posts in a variety of designs are fitted to suit the numerous designs of cutters, employed both for roughing and finishing. A square turret on the front of the slide to hold our tools is a common equipment.

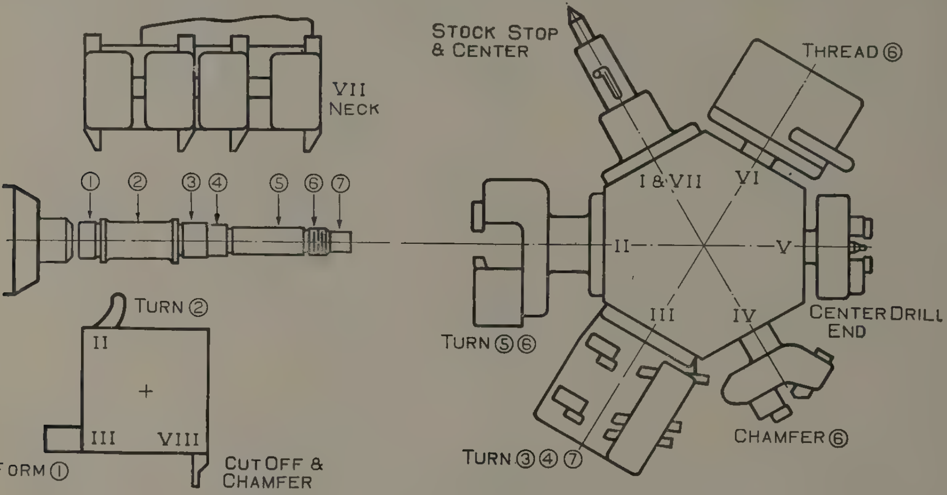


Fig. 144. Tooling Lay-out for Fan Shaft

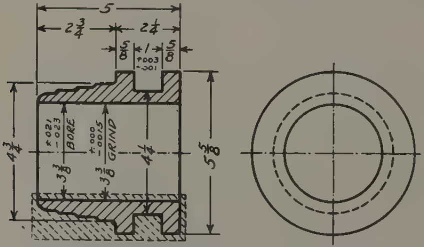


Fig. 145. Wedge or Cone for Operating Chuck Toggles

The Tools. Among the multifarious tools used, some types include many variations. Very briefly they are the following: The stop, set for arresting the projection of the stock bar at a precise length before commencing operations. Tools for centring, often combined with facing cutters. Pointing tools that chamfer the end of a bar before commencing to turn it. Turning tools have very numerous forms. The simplest are hollow mills, Figs. 139, 140—the first being of one dimension, the second having slight adjustability with the encircling ring. More elaborate forms have one or more cutters in a holder, working in opposition to a steady or steadies—either a vee, or rollers, embracing the work in opposition to the tools, and radially adjustable to size the diameter. These are the "box tools" which will often be turning more than one diameter, shouldering, and roughing and finishing. One of the simpler box tools is shown by Fig. 141—a Brown & Sharpe design. The shank fits in one of the holes in the turret. The rollers are supported centrally on pins that pass through the forks of slotted slides. The slides are adjusted radially with the screws *A*, *A*, and locked with the nuts *B*, *B*, on studs tapped into the slide. The cutters are supported on convex pads *D* for their tangential setting to the work, adjusted and clamped with the two screws above. A second holder is adjusted along the box body to deal with shoulders or ends or other operations.

Drilling and boring are done from the turret, either separately or at the same time as turning. Here, as in other cases, the operation that takes the longest time sets the pace, so that the other involves no expenditure of time. Flat drills are used for brass work, twist drills for other materials. They are clamped in sockets or in self-centring collet chucks. Reamers of all classes are used, solid-shanked, shell, expanding, parallel, and tapered. They are usually held in floating holders, the slight freedom of movement of which enables the reamer to follow its hole and smooth it. Boring tools are similar to those used in lathes and other machines. They are single, or double-cutting, inserted in bars with shanks, fitting turret holes, and, when the overhang is considerable, having pilots entering the spindle or a bush in the spindle. Minute adjustments for radius are made with screws.

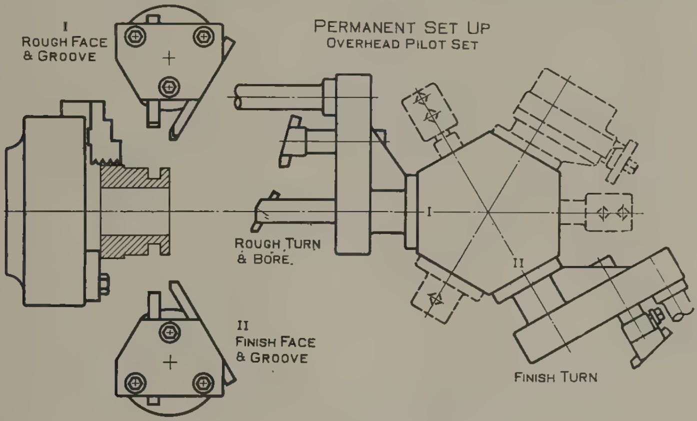


FIG. 146. TOOLING LAYOUT FOR WEDGE FIRST CHUCKING

Screw cutting is performed with solid, or with opening dies, or with chasers controlled with a master screw in front of the bed, sliding a chasing saddle. The rotation of the work has to be reversed when a solid die is run off the thread, or a solid tap from its hole. Opening dies are now in the ascendant, the chasers being thrown off contact automatically on the completion of a thread. Collapsing taps are also used to a lesser extent. The mechanism cannot be included in sizes less than about 1 in. An advantage which these and the opening dies have over the solid forms is that they can be adjusted to cut slightly under or over exact sizes.

The Work Done. The turret lathe is a machine of general utility, adaptable to small and large quantities—a dozen pieces, or ten thousand. For the first, standard tools are selected, for the second the entire set-up may be schemed anew with the object of saving seconds on a piece, when the high cost of the tool outlay shrinks into a matter of microscopic interest. The following drawings of sets-up, for which I am indebted to the Warner & Swasey Company, are self-explanatory. They show the piece of work, the batteries of tools, and the sequence of operations.

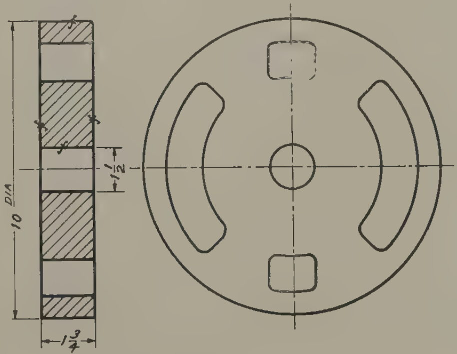


FIG. 147. CLUTCH RING

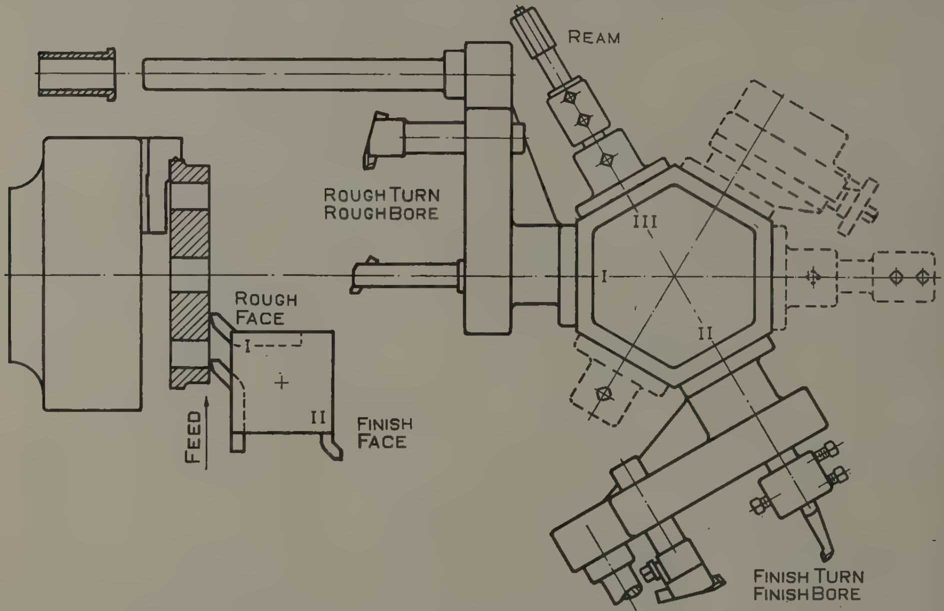


Fig. 148. Tooling Layout for Clutch Ring

Fig. 142 illustrates multiple cutting on a worm gear blank. It requires two outside turning, and one boring cut. A multiple turning head holds two cutters for the outside diameters, and a boring cutter for the hole, adjustable, and all cutting simultaneously. Fig. 143 shows a fan shaft, and Fig. 144 the tool layout for it. Six sets are mounted on the turret and two sets on the cross slide. Fig. 145 shows a chucking cone or wedge, and Fig. 146 its tool equipment. Multiple cuts are taken from the turret, and from the cross slide at one time. Fig. 147 is a clutch ring, Fig. 148 its tools. Facing is done from the square turret on the cross slide, turning and boring from the turret. Three of the turret holes are not required. The turning being heavy, the holder is steadied with a pilot bar.

BLAST FURNACE PRACTICE

By Percy C. R. Kingscott, D.I.C., A.R.C.Sc., F.I.C., B.Sc. (Lond.)

LESSON XI

FUELS

Fuels used in iron smelting are three in number, viz., coke, charcoal, and anthracite coal, of which the first is most widely used. Its predominance in this respect is becoming greater, and charcoal and coal are now used in limited cases. The duty of the fuel is twofold; it supplies heat and acts as a reducing agent towards the oxides of various elements present in the burden. As the rate of smelting is proportional to the development of heat, it is a *sine qua non* that the fuel should be readily consumed by the blast.

Coke

Coke is a light, porous fuel produced by the distillation of bituminous coking coals, and consists essentially of fixed carbon and ash. It contains also a certain amount of volatile matter residual from the coking process, some sulphur, and varying amounts of free water due to quenching the hot coke. In coking, the coal softens, swells, and evolves gases from this viscous mass, leaving a hard porous structure. Good coke is gray in colour, porous, yet hard and resisting. These qualities largely depend upon the coal, its preparation before coking, upon the temperature and duration of the process, and upon oven design and practice.

The impurities in coal are sulphur and phosphorus. Part of the former is removed during the coking process, but the latter, which is in the form usually of calcium phosphate, remains unchanged, being obviously of higher percentage content in the coke than in the coal. Its sulphur content, varying from 0·4 to 2 per cent, or over in inferior qualities, is the worst chemical feature of coke as a fuel. Generally, sulphur ranges 0·8 to 1·5 per cent. The amount of phosphorus present is generally fairly low, but this is very important in the manufacture of special low phosphorus pig-iron, as all the phosphorus present in the burden enters the metal, whereas sulphur can be dealt with by the proper use of slag making materials, and by hearth heat.

Coke ash is usually siliceous, and needs the addition of basic flux. The greater quantity of slag requires extra carbon to melt it, besides the fact that ash takes the place of carbon in the coke. Dirty coals usually receive a preliminary treatment which removes, in various ways, the inorganic matter not intimately associated with the coal substance. The ash in coke consists of free shale particles, natural ash and that carried over by the washery water in the form of fine slurry. This latter increases the sulphur content of the coke, and interferes with the cohesion of the particles of coal during the coking process.

Properties of Coke. Blast furnace coke is at present the cause of much discussion as to its essential properties regarded in the light of an efficient and economical smelting fuel. The chief characteristics are briefly described below—

1. **Porosity** is calculated from determinations of the true and apparent specific gravity. Porosity is undoubtedly a factor in the combustibility of coke but, with different cokes, the size of the pores and the thickness of the walls differ greatly, thus probably masking the effects.

2. **Size.** The general tendency nowadays is to use coke in pieces graded to pass a mesh of, say, 4 in. The combustibility of the fuel which governs the speed of the furnace is affected by the size of the fuel which, in turn, decides the surface presented to the oxygen of the blast. Modern American practice depends upon the rapid production of CO, and this is largely brought about by the grading of the coke, in order to achieve the greatest action with the blast. In addition, by reason of the furnace lines, the utmost freedom is given to the coke to reach the tuyeres (*J.I.S.I.*, Vol. CXII, 349).

3. **Resistance to Crushing.** Much stress has been laid upon this condition, but most normal cokes are capable of resisting compression in the furnace. Weill (*R. Met.*, 1905, 557) showed that breakage of coke in the blast furnace was not due to crushing, and Johnson has shown that whereas the pressure of the stock column never exceeds 35 or 40 lb. per sq. in., yet very ordinary coke withstands crushing stresses of several hundred pounds per square inch.

4. **Resistance to Abrasion.** A coke in its passage through the furnace must not produce

breeze, and must necessarily be strong enough to reach the tuyeres intact. Coke is tested in a rotary drum, which revolves a definite number of times, and the resultant coke remaining on a screen of definite mesh gives the hardness which is also connected with—

5. **Resistance to Impact.** This also concerns the production of breeze which embraces sizes from dust to pieces passing a 1-in. mesh. Breeze acts as a neutral body in the furnace instead of producing heat, and also is a fruitful source of furnace stoppage. It has been proved that the elimination of breeze by passing coke over a screen with 1¼ in. mesh results in a reduction of coke actually consumed. Sometimes this breeze is re-screened, and pieces over ½ in. in size are charged with the coke into the furnace.

Coke makers endeavour to produce a resistant coke, and modern furnace plant includes coke screens and charging gear whereby the number and extent of the coke drops are as low as possible. Some cokes are soft, yet resist "shattering" blows, whilst others withstand abrasion, yet are brittle owing to cracks possibly caused by over-coking or in the quenching process.

6. **Combustibility** is a measure of the reaction between the coke and the oxygen of the blast.

Combustible coke burns rapidly at the tuyeres, an intense hearth temperature thus ensuing. Arend and Wagner (*Re. de Met.*, Vol. 21, 585) state that a coke with a high combustion temperature, but not necessarily readily combustible, is required. But an insufficiently combustible coke causes combustion to extend too high in the furnace, causing high blast pressures and heat losses. Weyman (*J.S.C.I.*, Vol. 44) suggests that as coke is not a homogeneous substance it undergoes preferential combustion, and there is a range of temperature over which ignition is spread.

As was pointed out earlier, the coke sizes, combined with the furnace lines, have an important influence upon rate of burning. It seems to be agreed that the best blast furnace coke is made in narrow ovens, particularly where the coal has been cleaned and finely ground.

7. **Reactivity** has been defined as the equilibrium law of the reversible equation

$$CO_2 + C = 2CO.$$

Arend and Wagner (*loc. cit.*) show that reactivity increases with porosity, but cokes with equal porosity and made from different coals vary in activity, which thus depends upon chemical composition and coking methods. Reactivity increases with rise in temperature, and it is suggested that a saving in coke can be made by using a fuel with lower reactivity in the range 500–1,100° C., i.e. a hard and compact coke. It is possible to produce a coke readily burned by the blast yet with low reactivity in the reducing zones.

In general, under-coked and soft cokes are dissolved by CO_2, and tend never to reach the hearth.

Brassert has stated that the silvery graphitic skin on the coke pores renders the fuel less soluble in CO_2, yet affording rapid combustion at the tuyeres. Schreiber (*Brennstoff-Chemie*, 1923, Vol. 4) cites the work of Bone on decomposition of CH_4, and states that graphite is formed in this manner. Korevaar (*Combustion in the Blast Furnace*, p. 168) controverts this theory by stating that graphite soils the hand, whereas coke does not, moreover that graphite is not produced at coke oven temperatures. Mathesius (*A.I.S.I.*, May, 1917) admits that the ideal of a highly combustible coke immune from solution loss is apparently paradoxical, but claims that better coke practice has been obtained by the use of such fuel.

The influences of combustibility and reactivity which are intimately bound up with one another are thus not clearly settled, and it is possible that the good effects of modern fuel are entirely due to the production of hard tough coke in narrow ovens with short coking periods, and where the coals have been properly cleaned, mixed, and crushed before coking.

PRODUCTION

Coke is made in beehive and by-product ovens, the latter producing, nowadays, the majority of the coke output. The former are rapidly being displaced.

Beehive Oven. This oven, as its name implies, resembles in shape a beehive, being a section of a vertical cylinder about 13 ft. in diameter, about 3 ft. high and surmounted by a section of dome, the total height being 6 or 7 ft. The ovens are built of firebrick, capable of withstanding heat and abrasion, and are built in rows. Alternate ovens are charged and drawn, the coal being charged to a depth of about 3 ft. by means of an opening in the crown of the dome, and coke being withdrawn by means of a door at the front of the oven.

After levelling the coal and partially closing the dome opening, the distillation commences by reason of the heat remaining in the oven and its surroundings. At the end of the coking period the contents of the oven are quenched with water, this cooling producing a columnar structure in the coke. By-products are wasted in this process, the waste gases being usually led to waste-heat boilers.

A typical example of beehive coke is—

Ash	Volatile	Sulphur	Moisture
9·0	·35	·8	2·0 per cent

By-product Ovens. These ovens are concerned with the recovery of the valuable by-products, viz., ammonia, benzol, tars, etc., as well as the surplus gas not required for heating the ovens. There are several well-known types, such as the Otto, Coppée, Koppers, Semet-Solvay, Piette, Simon-Carvé, and Wilputte.

Briefly, the ovens form a series of narrow coking chambers about 10 ft. high, 16 to 24 in. in width, and about 30 ft. long, provided with a door at each end. Coal is either admitted loose through holes in the roof, or is pushed into position in the oven in the form of a stamped cake. In this type of oven, the coal is more quickly coked than in the beehive oven, and little heat is lost as the coke is pushed out of the chamber before it is quenched. The oven walls contain flues and burners in which about 50 per cent of the treated gases are burnt, the products of combustion giving up their sensible heat to checker work in regenerators before passing up the chimney. Being heated from either side, the coke has a cleavage plane up the centre. It has a dark grey colour and is generally free from lustre, unlike the beehive product. A typical analysis of patent oven coke is—

Ash	Volatile	Sulphur	Moisture
10·0	1·5	1·2	5·0 per cent

VALUATION OF COKE

The value of a fuel is dependent upon its content of available carbon, this being the amount of carbon left after deducting that necessary to melt the slag produced from its ash, sulphur, and added flux. In order to estimate the limestone required, an analysis of the coke ash as well as its percentage amount should be considered, as some ashes are more siliceous than others. However, Forsythe considers it sufficient to estimate the approximate amount of flux by taking 2 lb. of limestone per pound of ash and 3½ lb. of limestone per pound of sulphur. Taking a coke of the following analysis, and assuming the sulphur entirely enters the slag—

Fixed carbon .	84 per cent
Ash . .	10
Sulphur . .	0·95

the amount of stone required will be—

$$(2 \times 10) + (3{\cdot}5 \times {\cdot}95) = 23{\cdot}33 \text{ per cent.}$$

Assuming that the limestone produces 55 per cent of its weight as slag, then per 100 parts of fuel, the slag will be—

$$10 + {\cdot}95 + (23{\cdot}33 \times {\cdot}55) = 23{\cdot}78 \text{ parts.}$$

As before, taking 25 per cent carbon to melt the slag requires 5·94 parts of carbon.

Hence, available carbon in the coke = 84 − 5·94 = 78·06.

When comparing the cash values of different cokes, it is evident that the cost of the coke and limestone per ton, and the percentage content of available carbon, will be required.

CHARCOAL

Charcoal results as the dry distillation of timber in closed retorts. An average analysis is as follows—

Fixed carbon	Volatile	Moisture	Ash	Phosphorus
86·50	7·0	5·0	1·0	·02

It is a very friable and porous fuel, but is the purest in use. As mentioned before, it has only a limited scope, and is used in the smelting of rich, low phosphorus ores, where such ores and timber are available. The fuel consumption in such furnaces is much lower than in coke furnaces, owing to the absence of sulphur which enables the smelting to be accomplished in the presence of siliceous, easily melted slags.

ANTHRACITE

This natural fuel possesses many of the characteristics required in blast furnace operation. It is high in fixed carbon, low in ash, sulphur, and phosphorus, and is physically strong. It is undesirable on account of the absence of cell structure, and its tendency to decrepitate when heated. It is generally used mixed with some coke, and its consumption is confined to its own locality of production. Such furnaces are provided with by-product recovery plants, when not open topped.

FLUXES

As previously stated, the duty of the flux is to render fluid the gangue of the ore and the fuel ash, and also to remove the major portion of the sulphur in the burden.

The cheapest form usually is limestone, but as it is difficult sometimes to obtain a high-grade quality, those containing varying amounts of magnesia are then used. Frequently, a mixture of dolomite and limestone forms the flux.

Limestone is carbonate of lime usually associated with small amounts of silica, etc. When heated, it breaks up according to the equation—

$$CaCO_3 = CaO + CO_2$$

This elimination of CO_2 can be performed outside the blast furnace, and it would appear advantageous to do so, particularly as CO_2 is evolved in that zone of the furnace where it can attack carbon. Bell showed that external calcination had no marked advantage in cost. It is probable that lime absorbs CO_2 in the furnace top, and the theory has been propounded that the evolution of CO_2 from limestone in the furnace prevents the accumulation of carbon, due to the catalytic action of fine ores, which would otherwise tend to block up that part of the stack. The endothermic reaction also tends to cool the gases and prevent premature combustion (*J.A.I.S.I.*, 1917, Mathesius, *Chemical Reactions of Iron Smelting*).

The presence of magnesia seems to have no appreciable effect on the action of the flux although, theoretically, each molecule of MgO has a chemical power 1·4 times as great as that of a molecule of CaO.

At the same time, dolomite has been used with great success in many places, e.g. Alabama.

The flux is best used when broken up to a regular size, generally such that it will pass through a mesh from 4 to 6 in. It is advisable, however, to screen out the dust.

Valuation. The available base or efficiency of a limestone is that portion left after the CO_2, acid components, and sufficient lime to flux the latter have been subtracted. For example, taking a limestone with the following analysis—

$CaCO_3$	SiO_2	Al_2O_3
96	3·5	·5

and assuming the ratio of bases to silica in the slag to be 1·3 : 1—

$96 \times {·56} = 53{·}76$ CaO present

$3{·}5 \times 1{·}3 = 4{·}55$ CaO needed to flux the Silica.

Available base $= 49{·}21$

OUTLINES OF LOCOMOTIVE ENGINEERING

By A. Morton Bell, O.B.E., M.I.Mech.E., M.I.Loco.E.

LESSON VIII

THE FRAMING

There are two distinct forms of framing employed for locomotives, "bar" and "plate." The very early locomotives were built with frames constructed of iron bars welded together, and cross-braced with stays bolted on. Although this form of framing has been retained and is in general practice with engines of American design, British manufacturers prefer a framing of plate (now made of rolled steel slabs), maintained at the proper distance apart by transverse members built up of plate and sections, or as a one-piece detail of cast steel. For the bar-framing, greater flexibility is claimed, and doubtless there is much to be said for it on this score, for plate-framed engines are not favourites on railways with poor permanent way, or in countries where severe cold produces a hard track devoid of resilience.

The illustration (Fig. 32) shows one of the framed plates for a 4–4–0 locomotive, with the horn blocks, or guides, for the axle-boxes, either riveted or bolted on. In either case, the rivets or bolts should be a driving-fit in the holes, as a safeguard against movement during service, for there is a great tendency for looseness developing in a structure constantly subject to shocks and vibrations.

The axle-boxes can be of solid gun-metal, or of iron or steel, with bronze bearings inserted; these latter are usually lined with a white metal alloy, which contributes to a smooth running bearing, and acts as a check to damage of the journal of the axle, should a box run hot.

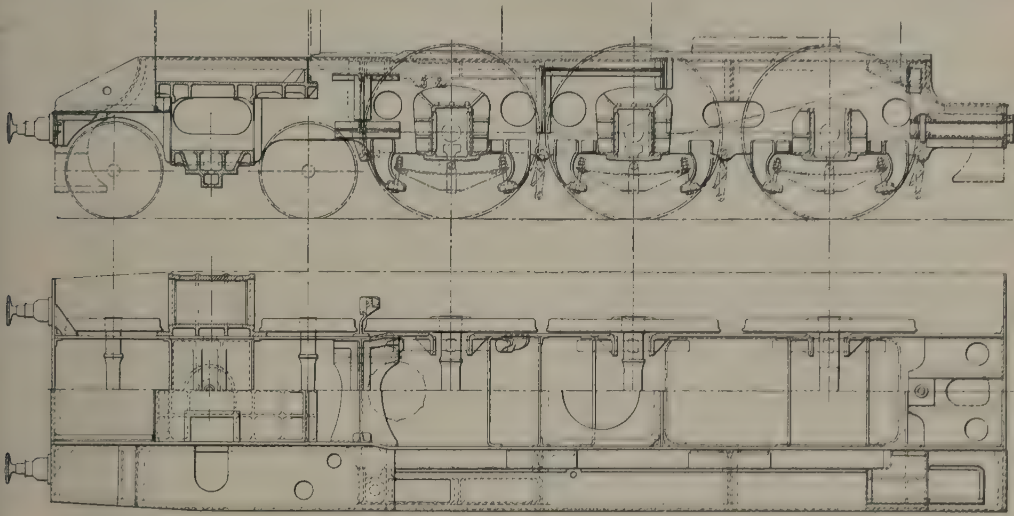


Fig. 32. Framing for 4–6–0 Locomotive

The axle-boxes are dropped over the journals, and maintained in position by a "keep," which forms an oil reservoir and receptacle for waste, or a pad, for lubrication purposes. The construction of a representative axle-box for use with under-hung springs is clearly shown in Fig. 33. To stiffen the guides and assist in maintaining parallel faces for the axle-box to slide up and down in, a stay, or strut, is provided, fitting closely between the legs of the guide.

A study of the framing shown in the drawing (Fig. 32) will enable the reader to follow the principles of the spring suspension adopted for the bulk of the engine and its superstructure;

the only parts which are not spring-borne, or cushioned, are the wheels and axles themselves, and it is therefore desirable to keep these as light as possible consistent with strength, service, and safety.

The springs are intended to soften, or absorb, the shocks received by the wheels when running over inequalities of the permanent way; they are of different forms, laminated, spiral, volute, etc. The first mentioned are made up of plates of spring steel about 5 in. wide by $\frac{1}{2}$ in., or $\frac{5}{8}$ in. in thickness. The plates are cut to lengths, so as to form gradually decreasing support to the main top or back plate; they are kept together by a band, or strap, shrunk on and locked with a bolt, or rivet, passed through.

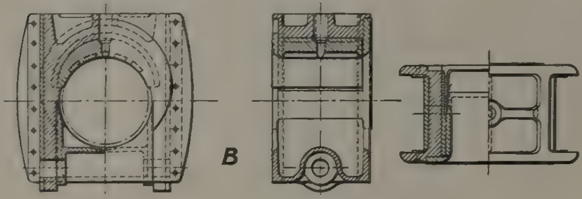


FIG. 33

There is often difficulty in finding sufficient room for suitable laminated springs under or above the driving axles, and, then, these have to be furnished with spiral, or volute, springs.

A good and much practised refinement is the provision of rubber blocks, to form auxiliary springs for the hangers, and deaden shocks transmitted to them; they are claimed to save the main laminated springs from much damage.

The bearing springs of the driving axles are coupled up with compensating beams, so that equalizing takes place throughout the system, to secure a good and equal distribution of weight over the different wheels of a locomotive.

The axles are invariably of steel, manufactured under stringent specifications for both method and material; the metal is subjected to severe tests. For outside cylindered engines, straight axles are used throughout, but for those having inside cylinders, it is necessary to have the driving axles "cranked" (see Fig. 34). These last-named may be forgings, worked up from ingots, or they may be built up from pieces machined and produced separately and then screwed, shrunk, or otherwise fastened together. The crank pins of a crank-axle are often drilled, and have bolts passed through to act as safety devices in case of breakage of the crank webs; the bolts also strengthen the make-up.

The wheel centres are either built up forgings of iron or are steel castings, more generally the latter, with spokes of oval section. They are provided with steel tyres, specially manufactured from hollow blooms by rolling, so that they present weldless circles of uniform quality. These are shrunk on to the wheel centres, and secured in many different ways against turning, the method adopted usually incorporating some interlocking device in the shape of a retaining ring, held in place by rivets, etc., so that should a fracture occur, the tyre cannot leave the wheel whilst this is running.

The driving wheels have balance weights, either fixed on or cast in them, to provide counter weights against the heavy reciprocating and revolving parts of the engine's mechanism. The bulk, or amount, of these weights is most carefully calculated, and often checked over by actual experiment, as it is very desirable to reduce to a minimum the "hammer-blow" on the permanent way, produced by the inertia of unbalanced weights.

It is usual to balance the total weight of the revolving parts and about two-thirds or three-fourths of those reciprocating. The "lead" given the slide valves assists in cushioning the latter members of the motion, but cannot, of course, reduce in any way the out-of-balance forces due to inertia.

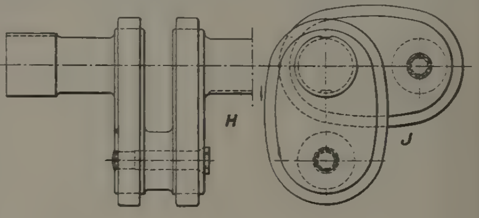


FIG. 34

With locomotives having coupled drivers, the weight of the reciprocating parts, is divided for balancing purposes over all the coupled wheels, but not necessarily in equal proportions.

To provide for necessary lateral flexibility in the wheel base of engines, those portions over and above that carried on the drivers are generally supported on a bogie truck of one, two, or in exceptional cases, three axles. The construction of a typical bogie is well shown in Fig. 35, and it is easy to imagine how its provision helps engines, such as those illustrated in Figs. 3 and 4, in passing round curves; without it, and all the wheels rigid, the flanges of the tyres at the ends of the framing would certainly bind hard on the rails on sharp curves.

The connection between the main frame of the engine and that of the bogie is usually made through a pivot and centre pin, which form the centre about which the bogie frame can swivel. The axles of the bogie have journals revolving in axle-boxes, carried in guides on the bogie frame, quite independent of the engine framing. The frames of the bogie are connected together by a central casting, in which the pivot carried by the engine is located. This pivot, in turn, is firmly attached to a casting, or cross-member, of the main frame of the engine. The latter is machined on its under surface, where it rests on the bogie casting. In addition to the rotating movement, allowed by the pivot, the top plate is permitted to slide laterally on the casting of the bogie under-frame, to the extent of 1 to 1½ in.; this movement is controlled by springs.

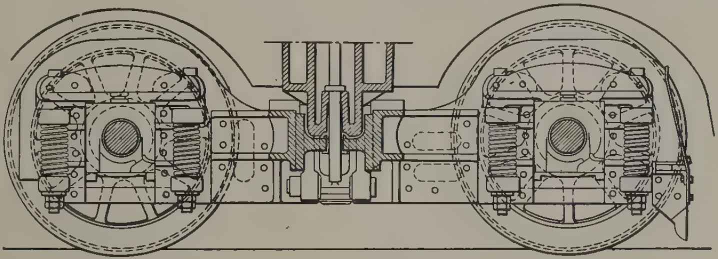


Fig. 35. Bogie

When the length of an engine precludes the use of a four-wheeled truck and something shorter, or smaller, would meet the weight to be carried, etc., a two-wheeled arrangement, or "pony," is used. There are very many methods of securing flexibility for such a pair of wheels; perhaps that originated by Bissell, in America, many years ago, is the most commonly used. Fig. 36 shows an adaptation of this, where the axle is carried by a triangular framing, pivoted to a cross-stay of the main framing of the engine.

There are also many forms of radial axle-boxes in use, depending on movement in curved guides, controlled by springs; these arrangements allow the axle to take positions radial with the curves of the line.

To the framing of the locomotive are attached the details used for coupling up the trains, including links, etc., for draft, and buffers for pushing on the British, or European, system, or central combined couplers and buffers on the American plan. Very considerable advance has been made in late years in the conversion from the former to the latter on foreign and colonial railways, and in this country the tendency towards a change is growing year by year, largely dictated by the need for increased speed in operation, the working of continuous brakes and electrification requirements.

The pull of the locomotive is transmitted through an elastic connection behind the draw hook of the tender, and this detail is rapidly passing from the elementary form it has hitherto assumed of plain volute springs or rubber pads to a much more elaborate friction draft gear, rendered necessary by the increasing weight of trains hauled. The side buffers are provided with steel springs of either volute or spiral form, compressed into the cases. In some buffers and draw gear, rubber blocks of special manufacture are used with very satisfactory results; they are utilized either independently or in combination with steel springs.

To meet the great length of many locomotives it is necessary for the buffer faces to be oval in form, or the frame of a long engine standing on a sharp curve may occupy such a position that its buffers will "lock-buffer" with those of an adjacent vehicle, if the heads of the buffers are not wide enough to prevent one getting behind the other.

An important detail on which the satisfactory adhesion of a locomotive, as well as assistance in retardation, is effected is the sanding gear; quite a number of different arrangements are in use.

Originally, a box of sand was carried on the locomotive and the contents sifted on to the rails by hand; next followed a pipe from the oot-plate down which it could be poured by hand. This latter particular method of sanding the rail is still practised on some mountain railways, due to the difficulty of applying it effectually by mechanical means on to the rails when passing round very sharp curves, etc.

Considering mechanical arrangements operated by steam or air, the sand which has been dried before storage, is drawn from the sand box by the suction of a steam or air jet, which then drives it under the wheels. It is not difficult to imagine, however, when a high side wind is blowing that sand may be scattered before it reaches the destined spot, consequently, apparatus has now been devised wherein a much more definite delivery is assured. The sand is moistened and stirred in the sand box before it is applied by the steam jet in the form of wet mud to the rails; this method has gained considerable application, and is said to most effectually secure the correct delivery of the grit, so much desired.

It has already been stated that the effective adhesion of a locomotive can be increased by 15 per cent by sanding the rails, and there are many other conditions arising in ordinary traffic wherein the sand box becomes a most essential adjunct.

All locomotives, tender or tank, are now provided with brake gear, although in early days this was usually confined to tank engines and the tenders of others, due to the difficulty of connecting up the two sets of gearing. With continuous brakes, either air or vacuum, this is now easily provided for, and in other cases steam brakes are generally fitted.

A brake cylinder is fitted below the foot-plate, and its power is applied to a cross-shaft which has arms connected to pull-rods, which operate the cross-beams of the hangers carrying the brake blocks. The brake hangers are levers, depending from a top pin which forms a fulcrum, and the pull at the lower ends transmitted through the cross-beams force the blocks against the wheels.

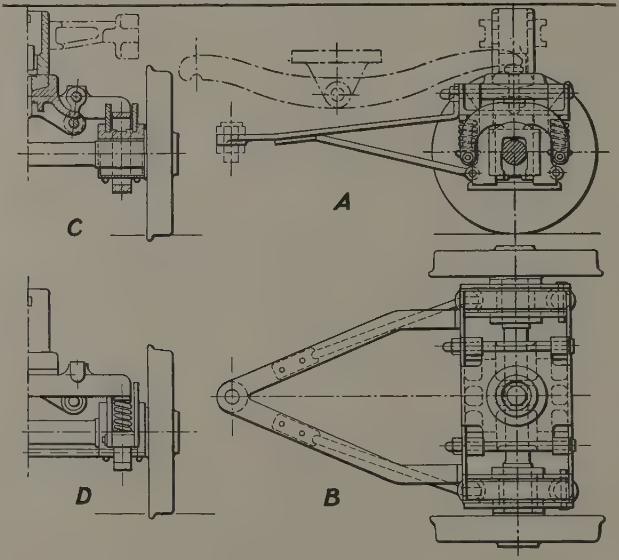


Fig. 36. Bissell Truck

A. Side elevation *B.* Plan *C.* Cross section *D.* End view

It is very desirable, wherever possible, to arrange the brake blocks so that they grip the wheels "crab-fashion," as this will obviate the very undesirable stresses produced by an unbalanced thrust against one side of the wheels only.

There is a growing tendency to fit the bogies, as well as the driving wheels and other carrying wheels, with brake blocks. This procedure is more easy of following when the air brake is used, owing to the small size of the cylinders required; if the steam brake is relied upon, then a complication ensues with the gearing unless a number of separate cylinders are used.

As the brake blocks wear, adjustment has to be made to take up the slack, or slow and irregular operation will result. It is also very important that the blocks should be well balanced on their hangers, or precautions taken against them tilting on to the wheels when the brakes are off. This tilting, or "hanging on," as it is termed, is often the equivalent in resistance to an additional coach to a passenger train. To prevent it, the blocks are often cast of such a form that the centre of gravity tends to keep them parallel with the wheels, but perhaps a more definite means is to provide springs, which, in turn, can be adjusted; without something of this nature there is always the loose play connected with such details to be accounted for.

Although pilots, or cow-catchers, are not deemed necessary in this country, they are very important fitments for locomotives running on railways unfenced and unprotected. In this country it is only customary to fit guard-irons on the front of the framing; these are placed directly over the rails, and are within 1 in. or so of touching them, so that the obstruction must be very small which could pass them; with bogie engines they should be attached to the bogie framing.

Finally, at the other end of the engine, there is the foot-plate, which usually consists of a casting, often purposely made somewhat heavy to counterbalance the weight of the cylinders, etc., at the front end. This casting, or "dragbox," as it is termed, carries the coupling attachments for connecting the engine and tender, and forms a cross-stay for the framing and a base for the cab; the platform on which the enginemen stand is built on it.

The cab which covers the foot-plate is now of much more commodious design than formerly, and although in the temperate climate of this country it is not considered necessary to completely close it in, back and front, the sides are brought well back and furnished with windows, which can be opened or shut by sliding. The front plate has as much window surface as possible, to permit of a good outlook being kept.

The arrangement of the various fittings requires careful planning, so that all the handles, valves, etc., shall be as accessible as possible, whilst the cab floor and fire door are kept clear for firing operations.

Fig. 36A shows the arrangement of the foot-plate of the G.W.R. 4–6–0 engine illustrated in Fig. 3. From this the position of the different fittings will be readily followed.

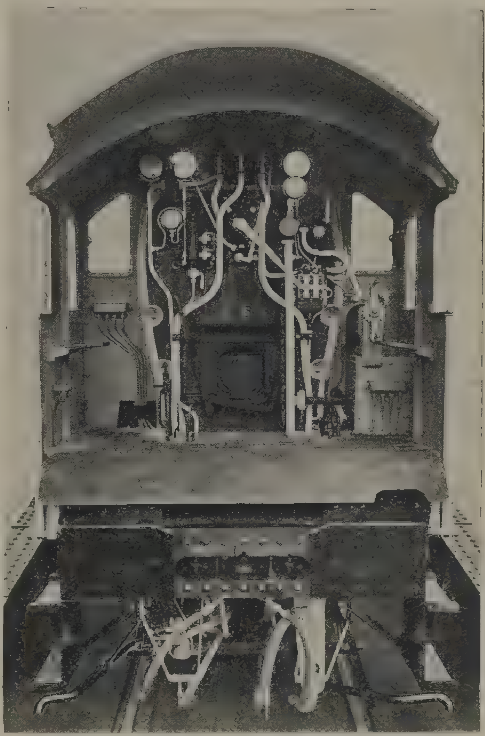


FIG. 36A. FOOT-PLATE, G.W.R. LOCOMOTIVE

In this country the correct position for the driver, that is to say, on the right-hand side or the left-hand side, has been much debated. On the engine shown the driver's position is on the right-hand, and it would certainly seem that this is the more convenient side for him to stand when firing is in progress.

The cab roof is prolonged, so as to shelter the front of the tender and protect the fireman from the weather.

THE BALANCING OF ENGINES

By E. B. Cole, B.Sc.

LESSON II

BALANCING OF REVOLVING MASSES—(*contd.*)

To draw the couple polygon—

A couple, as has been stated is a vector quantity, i.e. it can be wholly represented in magnitude and direction by a straight line.

The usual method of representing a couple is by a vector drawn perpendicular to the plane of the couple, i.e. parallel to the axis of the couple, the direction being that in which a right-handed screw would move if turned by the couple. Fig. 5 shows this convention, while in Fig. 6, on the same convention, the vector *ab* in the reference plane XY represents the couple Fx, normal to this plane, produced by the rotation of the load W, at the instant shown.

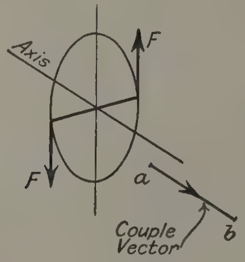


Fig. 5

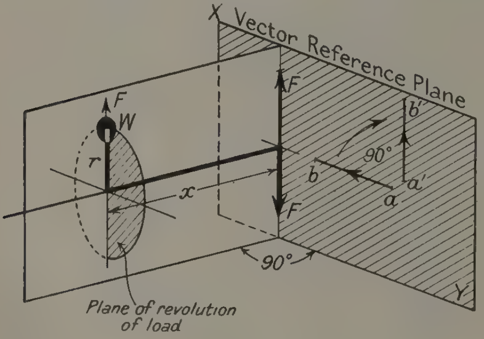


Fig. 6

But for our purpose it will be much more convenient if, instead of drawing each vector perpendicular to the plane of the couple, we turn it through 90 degrees, and make it *parallel to the radius arm "r" of its respective mass*, as indicated by a^1b^1. We can thus visualize the directions of the sides of the couple polygon without trouble.

Consider now Fig. 7, a force F_1 on the left of the reference plane, as shown, gives a clockwise moment about that plane, while a force F_2, acting in the same direction as F_1 but on the other side of the plane, gives a moment in the reverse direction.

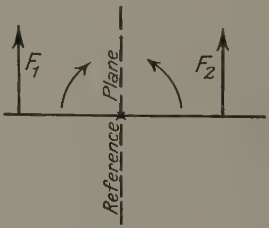


Fig. 7

Hence our rule for drawing the couple vectors can be stated as follows—

Choosing one side of the reference plane as positive and the other negative, couples on the positive side will be represented by a vector drawn parallel to the radius arm of the mass concerned in a direction measured from the axis outwards; those on the negative side, inwards towards the axis.

Fig. 8 shows pictorially a shaft subjected to three centrifugal forces, F_1, F_2, F_3, due to masses revolving in parallel planes, and their effect has been considered with regard to the parallel plane indicated.

1. The equal and parallel forces introduced in this plane have been combined into one resultant by means of the force polygon; and

2. The couples F_1x_1, etc., that occur normal to this plane have also been combined into a single resultant couple. Since the forces all act on one side of the plane, the couple vectors

are all drawn parallel to the radius arms from axis outwards.

In any given case, if the two polygons close, then there is no resultant force and no resultant couple occurring, i.e. the shaft is in perfect balance.

If the couple polygon closes for one position of the reference plane then it will close for all positions of this plane, provided the force polygon (which does not depend on the position of the plane) closes.

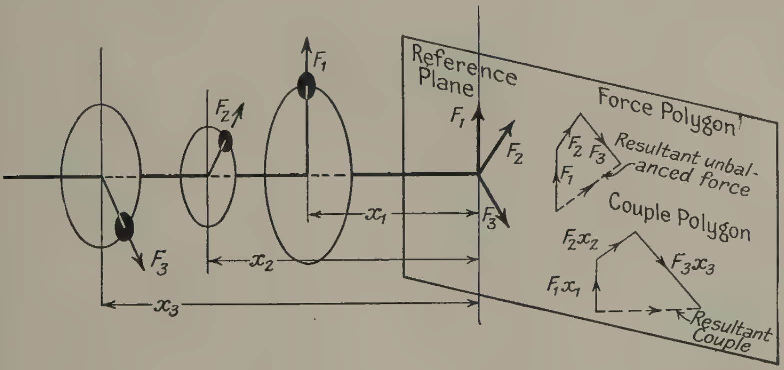


FIG. 8

Method of Balancing. If the closures of the two polygons happen to be *parallel*, then it is possible to effect balance of the shaft by the addition of a single mass, since its magnitude and radius arm may be so adjusted to set up the required balancing force, while its distance from the reference plane can also be arranged to give the necessary balancing couple, the two vectors acting in the same direction.

But, in general, the closures of the polygons will act in different directions, then it is possible to secure balance by the addition of *two* extra masses in separate planes, since these masses may be arranged to produce a resultant force in one direction and a resultant couple in another direction, thus closing the polygons.

Let us work out an example in detail.

EXAMPLE. Three masses of 500, 900, and 500 lb. rotate in planes whose relative positions are indicated in Fig. 9. It is required to balance this system by the addition of two masses, one each in the given planes *A* and *B*.

Determine the magnitudes and positions of these balancing masses.

Refer the forces to any reference plane, such as *R* (Fig. 9).

Now we have two unknown balance weights to find, which means that there are two unknown forces and two unknown couples—unknown both as regards magnitude and direction—and hence we have not sufficient information to complete either polygon, since an infinite number of figures can be drawn when two sides are completely unknown.

But if we choose as reference plane one of the planes containing an unknown balance weight (which we are at perfect liberty to do as any plane may be taken), we then eliminate one of the unknown couples since the required balance weight in this plane can exert no couple about this plane, and we can now draw the couple polygon.

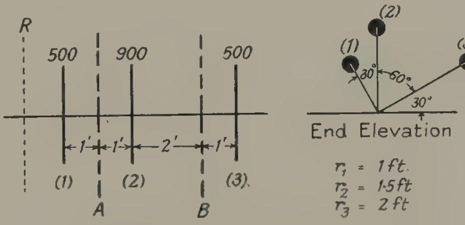


FIG. 9

The closing side of the couple polygon will represent in magnitude and direction the couple that must be exerted by the other unknown balance weight, which can therefore be estimated.

It is convenient to tabulate our information thus—

Reference Plane taken at *B*

Plane	W lb.	Radius r ft.	Distance from *R.P.* x ft.	Centrifugal Force $\propto Wr$	Centrifugal Couple $\propto Wrx$
A	730	(say) 2	3	1460	4380
B	830	(say) 2	0	1660	0
1	**500**	**1**	**4**	**500**	**2000**
2	**900**	**1·5**	**2**	**1350**	**2700**
3	**500**	**2**	**—1**	**1000**	**—1000**

(The figures in small type are the results obtained below)

From the last column it is seen that there is only one unknown couple—that which has to be supplied by the mass to be added in plane *A*. Drawing the *couple polygon*, the couples being taken in order in the usual manner, and that due to force in plane (3) being reckoned *negative*, the closure (dotted, Fig. 10 (*a*)) represents this couple and scales 4,380 units.

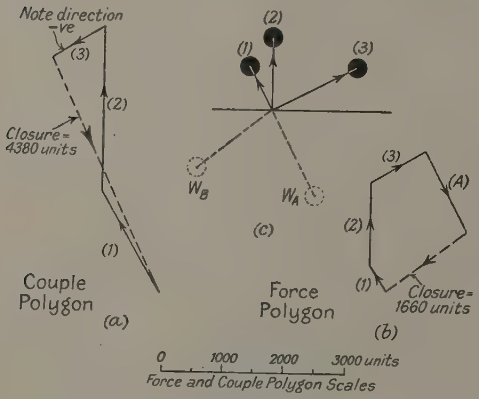


FIG. 10

$$W_A r_A x_A = 4380$$

But $$x_A = 3 \text{ ft.}$$

$$\therefore W_A r_A = 1460$$

Thus, taking $$r_A = \text{(say) } 2 \text{ ft.}$$

$$\therefore \underline{W_A = 730 \text{ lb.}}$$

Now the closure also gives the *direction* of the radius arm for this balance weight W_A, owing to our convention adopted for drawing the couple vectors.

We can now draw the *force polygon*, since there is only one unknown force—that required to be produced by balance weight *B*. The closure of the polygon, Fig. 8 (*b*), gives the magnitude and position of this balance weight, i.e.

$$\text{closure} = W_B r_B = 1660.$$

If r_B is taken = 2 ft.

$$\therefore \underline{W_B = 830 \text{ lb.}}$$

Thus, by the addition of masses of 730 and 830 lb. each at 2 ft. radius in the respective planes given, and in angular position on the shaft determined by the directions of the force and couple polygon closures, Fig. 10 (*c*), we have effected balance of the centrifugal forces acting and also of the couples they produce about the plane of reference *B*.

From our previous remarks, the shaft will now be seen to be in perfect balance, since, as the force polygon has been closed, the couple polygon, being closed for plane *B*, will close for any reference plane.

PUMPS: CONSTRUCTION AND MAINTENANCE

By Owen A. Price, M.I.Mech.E.

LESSON X

THEORY OF THE CENTRIFUGAL PUMP—(*contd.*)

In the previous lesson we have determined the total energy head which a forced vortex can possess due to the velocities of its particles, without reference to the manner in which these velocities are produced. It is not possible for any impeller, imparting energy by rotation, to *communicate* a greater head than this to the vortex, for the simple reason that the vortex cannot receive it. Therefore, in studying the transfer of energy to the water by an impeller, we must distinguish between the energy which the impeller seems *capable of transmitting* and the energy which the water can acquire. In this way we shall adhere to physical facts and avoid expressions which cannot be reconciled with the performance of any centrifugal pump.

Work Done During Outward Flow Through a Forced Vortex

Work Done per lb. of Water. In order to maintain flow from the low pressure region, at the eye of the vortex, into the high pressure region at the circumference, *work* must be done by the impeller on the forced vortex. In transmitting energy by revolving vanes *rotating forces only can do work* since all radial forces are in equilibrium on account of symmetry. Therefore only tangential resistances (i.e. components of reactions resolved at right angles to the radius) can oppose the rotation of the impeller.

Using the notation of Fig. 36, each lb. of water delivered leaves the impeller periphery with a momentum $\frac{w_2}{g}$ lb.-ft. hence its *moment of momentum* is $r_2 \frac{w_2}{g}$ lb.-ft., which is the turning moment opposing the rotation of the impeller. Against this resistance the impeller rotates with an angular velocity of, say, ω radians per second.

Since, in rotational dynamics, *work done* is measured by the product of the turning moment and the angular displacement in radians, the work done per lb. of water delivered per second is

$$\omega r_2 \frac{w_2}{g} = \frac{w_2 u_2}{g} \text{ ft.-lb. per lb. of water.}$$

This quantity represents the *virtual work done*, and may be either more or less than the vortex can absorb.

The virtual work done at different rates of flow will vary according to the outlet vane angle, and may be investigated in a manner similar to that adopted for Fig. 37, when we were considering the total energy head in a forced vortex. Thus, by assuming, as before, a constant peripheral velocity u_2 the value of w_2 may be calculated at various radial velocities for each of the five vane tip angles in turn and the virtual work done per lb. of water thus obtained, as shown in Fig. 38.

The significance of studying the work done *per unit mass* of water is that this quantity is the virtual height to which each lb. of water would be raised, or in other words the *virtual head input*, and we are enabled to make comparisons with the total energy head in the forced vortex. For instance, with no radial flow through the impeller, $w_2 = u_2$, and therefore the virtual head input $\frac{w_2 u_2}{g}$ becomes $\frac{w_2^2}{g}$ which, from Lesson IX (p. 1331), is identical with the total energy head possessed by a forced vortex under similar conditions. For this reason Figs. 37 and 38 are plotted to the same scale, and the total head (H_t) curves of Fig. 37 have been repeated as dotted lines on Fig. 38.

The combined diagram shows the *virtual* and the *sensible* head input, and therefore shows simultaneously the virtual and sensible work done per lb. of water.

Hence we may write the *sensible work done* on the water as

$$\left(\frac{w_2^2}{2g} + \frac{V_2^2}{2g}\right) = H_t \text{ ft.-lb. per lb. of water}$$

with the acknowledgment that whatever may be the virtual work done this expression covers the *maximum work which can be done on the vortex* regardless of whether or not the impeller is capable of performing the work.

Total Work Done. The total work done on the water passing through a forced vortex is the product of the number of lb. passed and the work done per lb. Thus

Total virtual work done

$$= n \times f_2 \times \frac{w_2 u_2}{g} \text{ ft.-lb. per second.}$$

Total sensible work done

$$= n \times f_2 \left(\frac{w_2^2}{2g} + \frac{V_2^2}{2g}\right) \text{ft.-lb. per second}$$

where n represents the number of lb. of water corresponding to unit radial velocity f_2 ft. per second. These two expressions have been graphed for different rates of flow in Figs. 39 and 40, and on Fig. 40 portions of the curves of Fig. 39 have been repeated as dotted lines.

Many facts, of the greatest importance, concerning the behaviour of centrifugal pumps, are completely elucidated by a careful study of Figs. 38 and 40 and all that they mean. In every case of energy transmission by mechanical means the virtual work must exceed the sensible work, hence it appears that with radial and forward bent vanes no rotating impeller whatever is capable of transmitting the appropriate total head H_t to the water, that is, the impeller cannot fully energize its corresponding forced vortex. Also, with backward bent vanes an impeller can fully energize its vortex within a range of radial velocities determined by the vane tip angle.

Qualifications to be Applied to the Hollow Forced Vortex. We have seen in Lesson IX that for a given peripheral velocity the lateral pressure at the outer circumference of a hollow forced vortex is less than that of a solid forced vortex by an amount equal to the lateral pressure of a smaller vortex equal in size to the core and

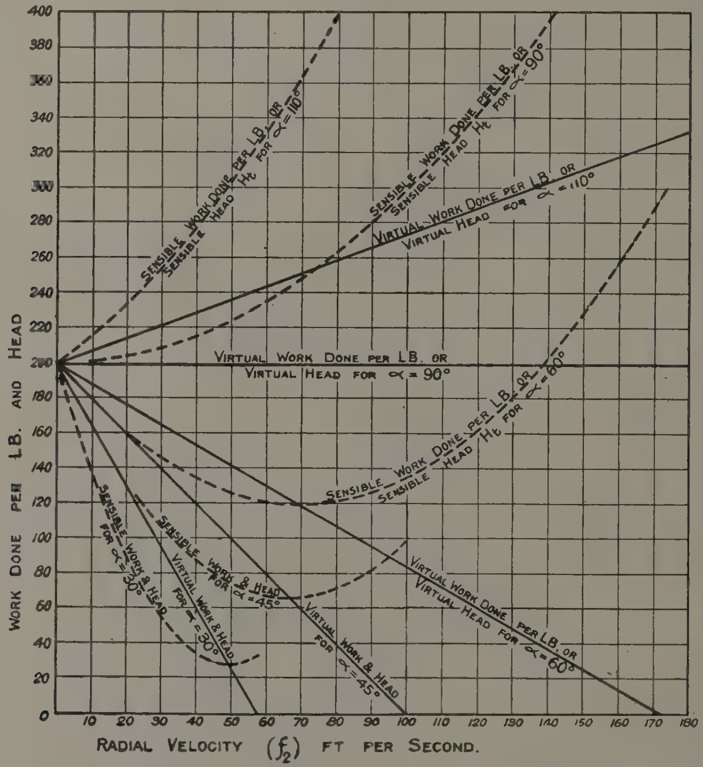


Fig. 38. Virtual and Sensible Work Done per lb., or Virtual and Sensible Head with Radial Flow through a Forced Vortex

rotating at the same angular velocity. Therefore the total energy of any particle in a hollow vortex differs from that of a particle in a solid vortex in the respect that the kinetic energy is in excess of the potential energy. Evidently, when there is no flow through the vortex, the conditions at the periphery are—

Centrifugal pressure

$$= \left(1 - \frac{r_1^2}{r_2^2}\right) \frac{w_2^2}{2g} \text{ ft. head.}$$

Kinetic energy

$$= \frac{w_2^2}{2g} \text{ ft. head.}$$

Total sensible head

$$= \left(2 - \frac{r_1^2}{r_2^2}\right) \frac{w_2^2}{2g} \text{ ft.}$$

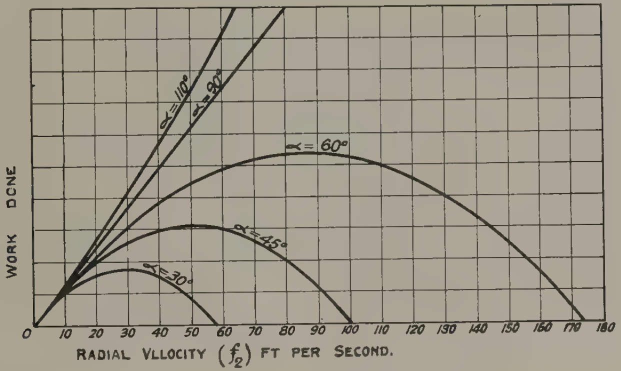


FIG. 39. TOTAL VIRTUAL WORK DONE

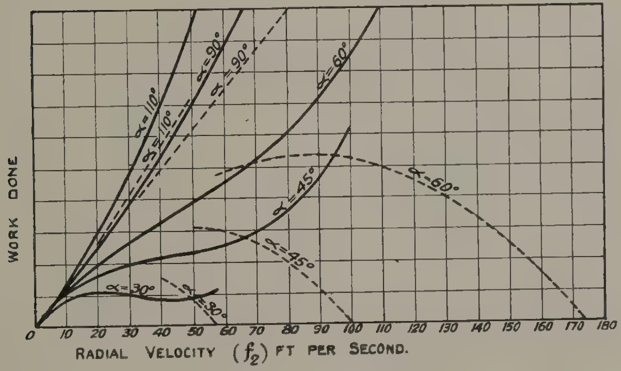


FIG. 40. TOTAL SENSIBLE WORK DONE—DOTTED LINES SHOW VIRTUAL WORK DONE

and whilst flow is in progress—

Centrifugal pressure

$$= \left(1 - \frac{r_1^2}{r_2^2}\right) \frac{w_2^2}{2g} \text{ ft. head.}$$

Kinetic energy

$$= \frac{V_2^2}{2g} = \frac{w_2^2}{2g} + \frac{f_2^2}{2g} \text{ ft. head.}$$

Total sensible head

$$= \left(1 - \frac{r_1^2}{r_2^2}\right) \frac{w_2^2}{2g} + \frac{V_2^2}{2g} \text{ ft.}$$

or

$$= \left(2 - \frac{r_1^2}{r_2^2}\right) \frac{w_2^2}{2g} + \frac{f_2^2}{2g} \text{ ft.}$$

therefore the total work which can be done on the hollow forced vortex is

Sensible work done

$$= n \times f_2 \left\{ \left(2 - \frac{r_1^2}{r_2^2}\right) \frac{w_2^2}{2g} + \frac{f_2^2}{2g} \right. \text{ ft.-lb.}$$

$$= n \times f_2 \times H_t \text{ ft.-lb.,}$$

or, expressed in horse-power

Sensible water horse-power

$$= \frac{n \times f_2 \times H_t}{550}$$

The notation used in the above expressions is identical with that used throughout our discussion, but for ready reference it has been repeated in Fig. 41, which shows the usual velocity diagrams for flow through a centrifugal pump.

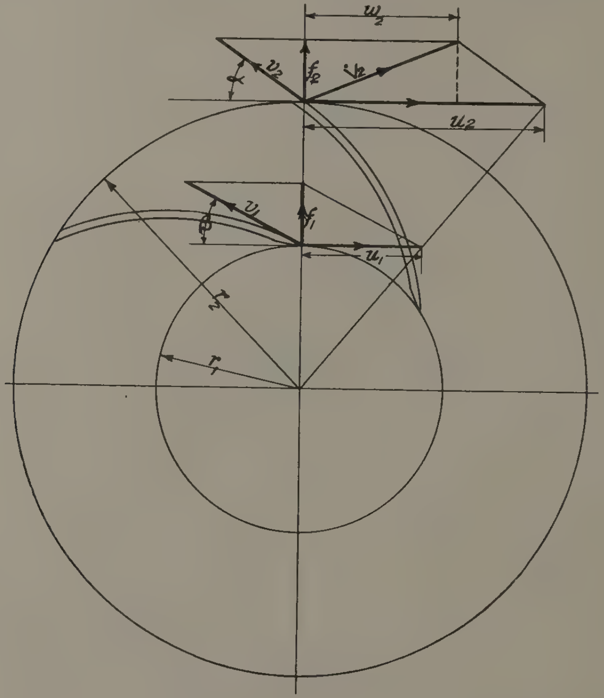


FIG. 41. CENTRIFUGAL PUMP VELOCITY DIAGRAMS

Useful Output of a Centrifugal Pump. The useful head developed by a centrifugal pump is *head output* = centrifugal pressure + C (kinetic head) where C represents the proportion of the kinetic head which is recovered. The value of C varies from practically zero, when there is no flow through the pump, to 0·8 or more on a well-designed pump of large size when discharging its normal output. The loss of kinetic head is due partly to an abrupt change of velocity as the water passes from the rotating impeller into the fixed casing, and partly to the loss in the divergent channel (if there is one) leading from the collecting passage to the delivery branch of the pump, or, in the case of a multi-stage pump, to the exit from each stage. Both types of loss have been dealt with in Lesson VIII, the former loss being approximately that of a sudden enlargement, and the latter loss that occurring in a divergent passage with gradually enlarging area.

The amount of kinetic head recovered is the principal factor in determining the characteristic shape of the head output curve of a centrifugal pump with backward bent impeller vanes, and the sensible water horse-power curve determines the general shape of the curve of power input necessary to drive the pump. Fig. 42 has been drawn for an actual test of a well-designed pump to illustrate the formation of the performance curves of a centrifugal pump. The essential particulars and data for this pump and test are

Peripheral velocity of impeller tip

$$= u_2 = 37{\cdot}15 \text{ ft. per second.}$$

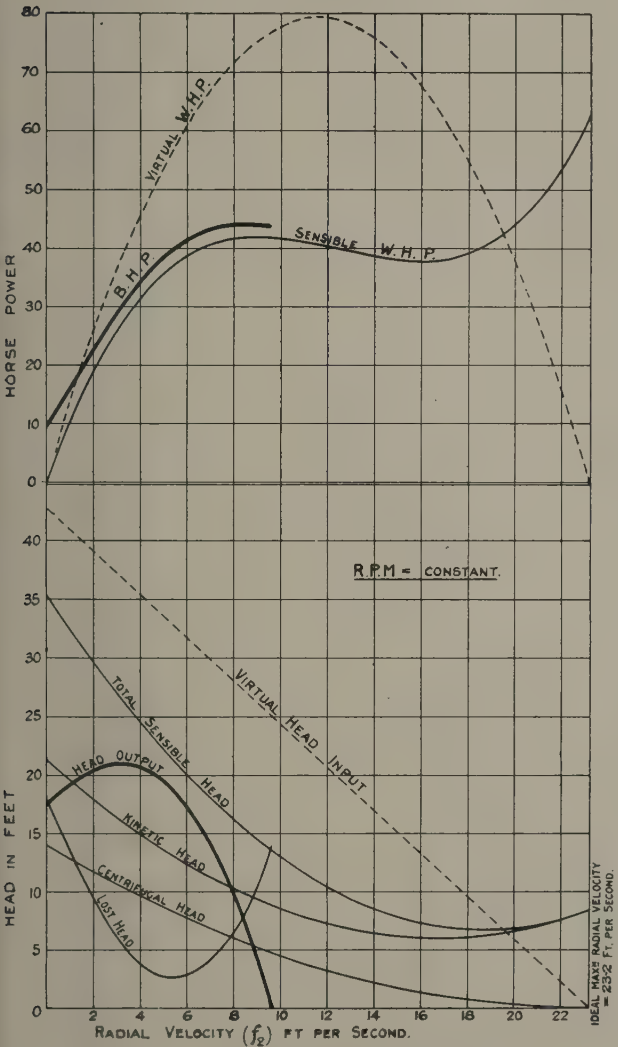


FIG. 42. CENTRIFUGAL PUMP BEHAVIOUR

Outlet vane tip angle

$$\doteq \alpha = 32^\circ \text{ with tangent.}$$

$\therefore$ Ideal maximum radial velocity

$$= 37{\cdot}15 \tan 32^\circ \doteq 23{\cdot}2 \text{ ft. per second.}$$

Velocity of whirl

$$= w_2 = u_2 - f_2 \cot 32^\circ$$

$$= (37{\cdot}15 - 1{\cdot}6003 f_2) \text{ ft. per second.}$$

$$\text{Ratio } \frac{r_1}{r_2} = 0{\cdot}5825. \quad \therefore \left(1 - \frac{r_1^2}{r_2^2}\right) = 0{\cdot}66.$$

Quantity of water passing per second per unit radial velocity $= n =$ 175·6 lb.

The various curves are calculated by the expressions given for the hollow vortex, and are similar generally to those of Figs. 37, 38, 39, and 40. The head output and brake horse-power are taken from test results.

The measured efficiency, during test, of the pump in question was over 80 per cent, the hydraulic efficiency shown by the diagram is about 88 per cent, and the mechanical efficiency therefore about 91 per cent, which is a reasonable agreement with expectations.

The virtual head input and virtual water horse-power are dotted in as a warning to show that these quantities have no relation to the useful performance of an actual pump.

Points in Design. Hitherto attention has been devoted to the conditions which exist at the rim of an impeller. The water velocity into the eye and entry between the inlet ends of the impeller vanes is determined by the maximum permissible vacuum at this point, when the pump is delivering at its utmost capacity and lifting against its maximum suction height. Though a low velocity is desirable, commercial necessity often requires high velocities. No rule can be given, 10 to 15 ft. per second is safe, though 20 to 25 ft. per second may also be safe under suitable conditions, and when a large suction pipe is employed on low suction lifts.

For the water to enter the vane passage with the minimum shock the inlet edge of the vane should be tangential to the line of motion of the vane tip relative to the entering water. Thus, in Fig. 41, the vane would be tangential to v_1, which is compounded of the circumferential tip velocity u_1 and the radial velocity f_1 of the water. If the inlet angle β is incorrect a loss occurs, due to the abrupt change of direction of flow (see Fig. 31C); however, great accuracy in determination of β is not of vital importance, a more important matter being that β should not be too small or throttling and turbulent flow will occur.

The most important problem in centrifugal pump design is the recovery of the kinetic head in the absolute stream of water leaving the impeller. In ordinary practice one of two methods is adopted, according to circumstances. When the volume of water to be pumped is large in comparison with the head to be overcome, the spiral casing, or volute (Fig. 26A), is employed, and when the head is large in comparison with the volume, the guide vane form of diffuser (Fig. 26B) is used. Many varieties and refinements of passage shape are used for both these types and in most cases a certain amount of free vortex chamber intervenes between the impeller and true water passage. Such short free vortex chambers are valuable in permitting unification of the several absolute streams escaping from the impeller, thereby reducing shock at entry to guide vanes or volute and invariably improving the range and efficiency of the pump.

STRUCTURAL STEELWORK

By W. Arnold Green, M.A., B.Sc., A.M.Inst.C.E., M.I.Struct.E.

LESSON VI

STRUCTURAL STEEL—(*contd.*)

Columns, Stanchions, Pillars, and Struts. Unlike tension members for which a fairly accurate estimate can be made of the breaking axial load, there is considerable uncertainty as to the load a compression member will carry. A slight departure from straightness of a tension member will tend to disappear under load, but in a compression member any such deflection will increase with the load. The fact that a tension member is rigidly held at the ends will not affect the axial load causing failure, but in a column it makes a great difference as also does the length of the member.

Volumes have been written on this difficult subject, and there are numberless formulae giving the crippling loads on struts; some purely empirical attempting to agree with the results of experiments, and some with a theoretical basis attempting to take into account inequalities in the elastic properties of the material, initial deflection, conditions of fixity at the ends, etc.

When a designer has chosen, or has specified for him the formula to be used, he is not concerned with its derivation.

The stiffness of a compression member is usually defined by the ratio of its length (l) to its least radius of gyration (k), where k is the square root of the minimum value of the fraction $\frac{I}{A}$,

where I = moment of inertia,
,, A = cross-sectional area.

Rankine's well-known formula for the crippling stress (fc) on a strut with *round* ends, i.e. with the ends unrestrained in direction, is in the form $fc = \dfrac{f_{max}}{1+a\left(\dfrac{l}{k}\right)^2}$. Commonly accepted values in this country for the constants for standard steel are 21 tons per sq. in. for f_{max}, and 1/7500 for a. By dividing by a suitable factor of safety (say 3) the safe stress is obtained.

As a strut with rounded ends is assumed to have no restraint at the ends, the failure may be expected to be by bending in the way indicated in Fig. 32(*a*).

If one end is fixed in direction, the column will bend as shown in Fig. 32(*b*), and if both ends are fixed, as in Fig. 32(*c*).

Thus, the strut which will have the same buckling load as the one with hinged ends is 50 per cent longer if one end is fixed, and 100 per cent longer if both ends are fixed.

As in practice a rigid connection at the end of a strut is rarely obtained, the late Professor T. C. Fidler, whose practical column formula has met with wide acceptance, recommended that a length one and one-third instead of one and a half times the length with hinged ends should be taken for one end fixed, and one and two-thirds instead of two times the length with hinged ends, if both ends are fixed.

Another common form for the formula for the allowable stress (f) in compression members

is $$f = f_1 - a_1\left(\frac{l}{k}\right)$$

Thus, in the formula for both ends fixed giving the stresses allowed by the regulations for steel frame buildings in London, $f_1 = 6\frac{1}{2}$ tons per sq. in. and $a_1 = 1/40$ up to a length equal to 160k. For greater lengths the constants are $f_1 = 10\frac{1}{2}$ and $a_1 = 1/20$. The allowable stress thus regularly reduces by 1 ton per sq. in. for every increase of 40k in the length up to 160k, and then by 1 ton per sq. in. for every increase of 20k.

It seems to be the practice in London to regard fixed ends as an unattainable ideal, and to use the stresses allowed for one end fixed and one end hinged, which are given by the formula in which $f_1 = 5\frac{1}{2}$ tons per sq. in. for lengths up to 140k and 9 tons per sq. in. thereafter, the value of a_1 being the same as for both ends fixed.

It will be noted that the allowable stress for this condition and for lengths up to 140k are 1 ton per sq. in. less than those for fixed ends for which there seems to be no theoretical justification, as for short struts the condition of fixity at the ends makes little difference to the crippling load. Nor is there theoretical justification for the more rapid falling off in the allowable stress for longer lengths, though this rule makes the use of slender struts very uneconomical.

The formula given in B.E.S.A., No. 153, is in the same form but with $f_1 = 8$ and $a_1 = 0{\cdot}0264$ for the stress in a compression member with rivetted connections at both ends, and $a_1 = 0{\cdot}04$ for pin connections.

In each case the stress must not exceed 6·8 tons per sq. in., nor must the length of the member exceed 120k or forty-five times the least width.

In both bridge and building construction, it is rare for a compression member to have a purely axial stress, though frequently with the stresses commonly specified it is safe to assume that they are axial.

When a girder is connected to the side of a stanchion and deflects with its load, the ends cannot be titled to the same extent as if they were free to move because the stanchion will tend to keep the ends horizontal, though not completely, as the rivets or bolts connecting the ends will stretch slightly and the stanchion itself will bend. Any bending moment at the ends of the girder decreasing the bending moment in the span must be shared between the stanchion lengths above and below the connection in a proportion depending on the moments of inertia, the lengths and the relative fixity of the other ends. In a frame such as that illustrated in Fig. 33, if the girder BE alone is loaded the bending moment diagrams and deflection curves for the beams and stanchions will be as indicated.

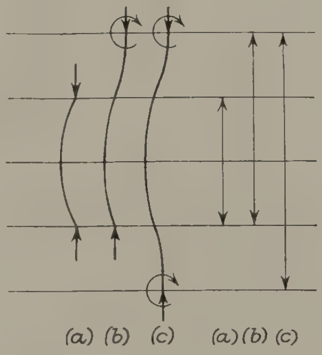


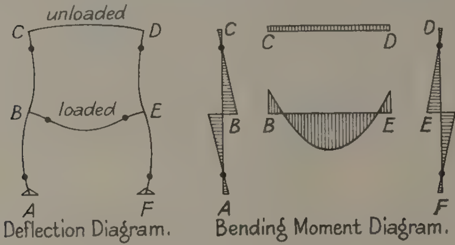
FIG. 32

As in the case of wind stresses, on the assumption of absolutely rigid joints the effect of the load on BC throughout the structure can be mathematically determined though the process of calculation, whether graphical or analytical, is somewhat tedious.

The process can be simplified by assumptions which sometimes may not seem very reasonable, but enable satisfactory structures to be designed.

As has already been pointed out, the design loads are often only vague guesses, so that elaborate calculations in such cases are not justified.

In steel frame buildings it is common to assume that the girders have no end fixity, and that the end reaction is at the centre of the seating. In the case of a girder connected to the flange of a stanchion, the centre of action is often taken as the edge of the flange and the corresponding bending moment instead of being shared between the column lengths above and below is assumed to be resisted by the lower length only.



Deflection Diagram. Bending Moment Diagram.

FIG. 33

It is commonly assumed that all loads from floors above are evenly distributed over the stanchion area at floor level.

Thus, if a stanchion AB has to carry an assumed central load P from all sources other than the girder BE, which brings a reaction on to the flange edge a distance n from the centroid of the stanchion whose area is A, radius of gyration parallel to the beam k_x, moment of inertia $A \cdot k_x^2$, and compression section modulus $A \cdot k_x^2 \div n$, then the axial load is $P + W$ combined with the bending moment $W \cdot n$. The stress due to axial load $= (P + W) \div A$; that due to bending is $W \cdot n^2 \div Ak_x^2 = \dfrac{W \cdot n^2}{k_x^2} \div A$.

The value of $\dfrac{n^2}{k_x^2}$ for rolled steel joists and plated sections is about one and a half, so that the *equivalent central load*, i.e. the central load producing the same stress as the load applied at the edge of the flange, is two and a half times the flange load.

Thus, if $P = 30$ tons and $W = 20$ tons, the load carried to the foundation will be 50 tons, but the stanchion itself will be selected from safe load tables for a central load of $50 + 1\frac{1}{2} \times 20 = 80$ tons.

Suppose that, for the length BC an 8 in. × 6 in. R.S.J. is suitable, and it is required to show calculations for the length AB (10 ft. long) for a plated 8 in. × 6 in. R.S.J., using the stress given by the formula $f = 5\frac{1}{2} - \dfrac{l}{40k}$. For an 8 in. × 6 in. R.S.J. with a 9 in. × $\frac{1}{2}$ in. plate each side, $A = 19{\cdot}3$ in.^{2}; $n = 4\frac{1}{2}$ in.; $k_x = 3{\cdot}79$ in.; $n^2/k_x^2 = 1{\cdot}4$ in.; $k_y = 2{\cdot}04$ in.; $\dfrac{l}{k} = 120/2{\cdot}04 = 59$; allowable stress $= 5{\cdot}5 - 1{\cdot}48 = 4{\cdot}02$ tons per sq. in.; allowable central load $= 19{\cdot}3 \times 4{\cdot}02 = 77{\cdot}6$ tons. Equivalent central load $= 50 + (1{\cdot}4 \times 20) = 78$ tons. Alternately, stress due to direct load $= 50 \div 19{\cdot}3 = 2{\cdot}59$ stress due to bending $= 20 \times 4\frac{1}{2} \div 61{\cdot}7 = 1{\cdot}46$ where $61{\cdot}7$ in.^{3} is the value of the section modulus.

Total flange stress $= 2{\cdot}59 + 1{\cdot}46 = 4{\cdot}05$ tons per in.^{2}. Further points in column design will be discussed in the next lesson.

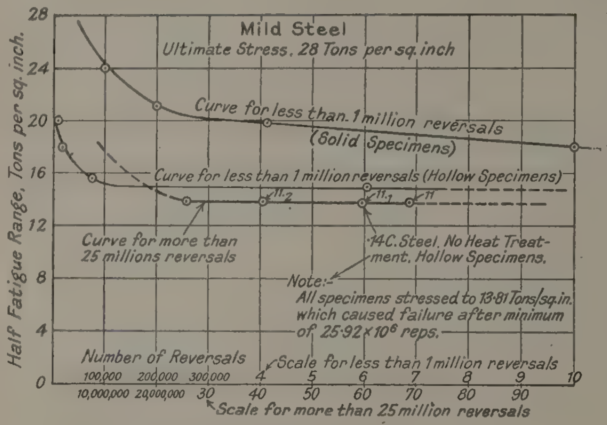


FIG. 17. STRESS REPETITION CURVES FOR MILD STEEL

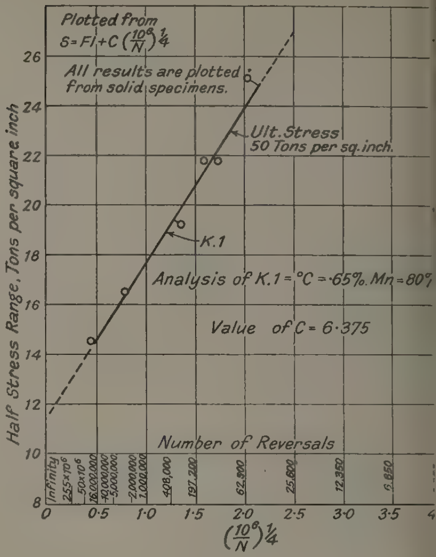


FIG. 18

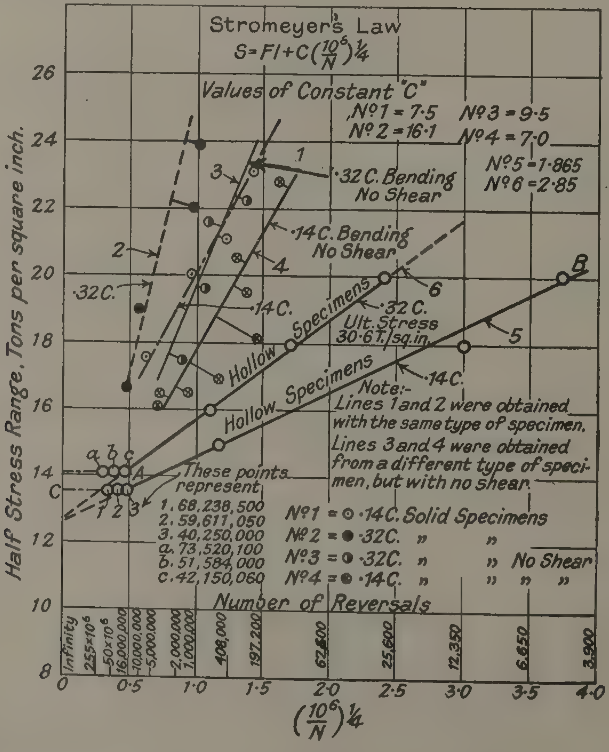


FIG. 19

REPEATED STRESSES IN MATERIALS

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PART II

RESULTS OF TESTS

(*Contd.*)

The simplest method of plotting the results of tests is to plot number of repetitions to cause fracture against range of stress as calculated from the applied loads. If the specimen is subjected to bending, the maximum stress can be calculated from the well-known formula

$$p = \frac{My}{I}$$

and if subjected to torque from

$$q = \frac{Tr}{I_p}$$

In both of these formulae it is assumed that the stress during the loading is proportional to the strain.

When subjected to direct stress, as in the Haigh machine, the pull or push is known from the calibration of the machine. For the calibration, an extensometer is used on a special specimen (Fig. 15). This specimen is loaded in a testing machine and a load strain curve taken. It is then put in the Haigh machine and subjected to repeated stresses. The mirror oscillates and the strain is measured by the movement of a spot of light on a ground glass scale. The leaf springs of the machine can also be calibrated by this extensometer.

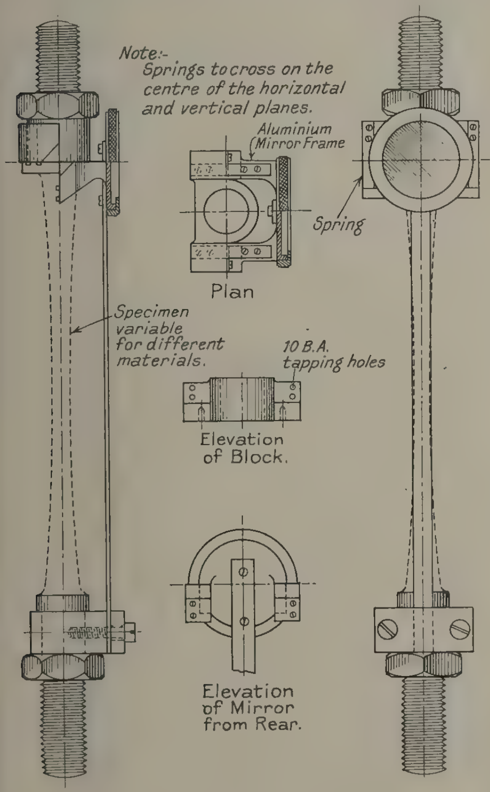
Note:-
Springs to cross on the centre of the horizontal and vertical planes.
Aluminium (Mirror Frame
Spring
Plan
Specimen variable for different materials.
10 B.A. tapping holes
Elevation of Block.
Elevation of Mirror from Rear.

Fig. 15. Mirror Extensometer

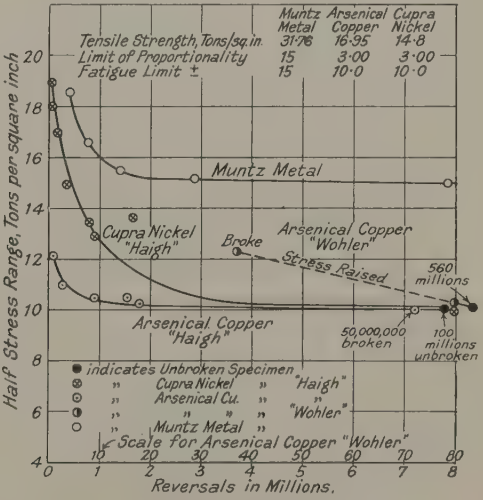

Fig. 16. Stress Repetition Curves for Various Metals

Tests at Zero Mean Stress. Figs. 16–17 show typical stress repetition curves. It will be seen that in all cases the curves tend to become parallel to the axis of time, or reversals, after a considerable number of repeated stresses.

Fatigue Limit. The important question is: "Do these curves really become parallel to the axis of time ? " If they do, then the stress at which they become parallel is the real fatigue limit, and at any stress less than this stress an infinite number of repetition stresses can be applied without risk of fracture. It is not easy to determine whether the curves do really become parallel to the axis of number of reversals.

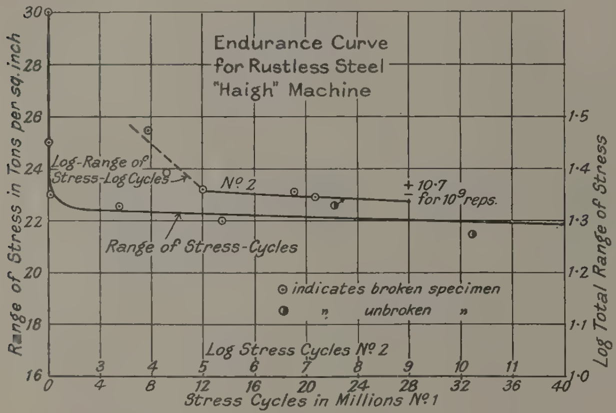

FIG. 20

Other methods of plotting have been therefore suggested. In Figs. 18 and 19 the results of tests have been plotted by taking as ordinates range of stress, and as abscissae $\left(\frac{10^6}{N}\right)^{\frac{1}{4}}$ where N is the number of repetitions. The zero abscissa then corresponds to N equal to infinity. If the points lie about a curve which appears to cut the vertical axis of the curve, then it may be assumed that the limiting range fatigue is equal to or less than the stress at which the curve cuts this axis. It will be seen for the cases plotted that the points lie about straight lines, which, when produced, cut the vertical axis. The author's tests on certain steels indicate quite clearly that for these materials the limiting fatigue range is above the point where the line cuts the axis. In mild steel the line appears to fall continuously until the number of repetitions is between 10 and 16 millions, and then turns horizontally to cut the vertical axis above the point at which the line through the points cuts the axis. It will be seen that the slopes of these lines vary with the material and with the type of specimen.

Logarithmic Plotting. Fig. 20 shows, in the upper curve, log. range of stress plotted against log. number of repetitions, as well as stress against number of repetitions. In nearly all cases of such plottings, the curve falls rapidly for the first five million repetitions, and then becomes either parallel to or nearly parallel to the axis of log. N. If it becomes parallel to this axis, then the limiting fatigue stress is

TABLE I

SAFE RANGE OF STRESS FOR MORE THAN 10 MILLION CYCLES AND EQUAL ± STRESSES

Material	Heat Treatment	Tensile Strength, Tons per Sq. In.	Safe Range for 10 million cycles. Tons per Sq. In.	Ratio of Range to Ultimate
·14 C.	Nor. (Solid)	30·6	± 15 to ± 16	1 say
·32 C.	,, ,,	36·42	± 17 to ± 18	1 ,,
·14 C.	No heat treatment from 2 in. Bar	27·98	± 14 to ± 14·5	1 ,,
·14 C.	Heat at 850° C. 36 hours	26·45	± 14	1·06 ,,
·14 C.	Heated at 1,100°C. 1 hour	23·99	± 12·7 to ± 13·5	1·06 ,,
·32 C.	,, ,,	28·40	± 12·5 to 13	·93 ,,
·32	Heated at 850° for 36 hours	32·01	± 15	·93 ,,
·14	Quenched	48·5	± 21·8	·88 ,,
·32	,,	89·9	± 37·1	·82 ,,
Alloy Steel	Quenched and Tempered	90	± 37	·82 ,,
,,	,, ,,	70	± 34	·91 ,,
,,	,, ,,	60	± 30	1 ,,
Muntz Metal	—	31·76	± 15	·95 ,,
" Cupro-Nickel "	—	16	± 5·1	·63 ,,
Arsenical Copper	—	14·8	± 5·1	·68 ,,
Rustless Steel	—	54	± 11	·40 ,,

found; if not, it is uncertain. With most materials, and particularly with steels, there seems no doubt that thousands of millions of repetitions of stress can be applied to the specimen without risk of fracture at a range not much less than that at which 10 million repetitions will just cause fracture. With heavily cold worked materials and for certain alloys the range seems to be continuously but very slowly falling up to 500 million or even 1,000 million repetitions, but in these cases the rate of fall beyond 10 million repetitions is small, and it can be stated, therefore, that *the safe range of repetition stress, for zero mean stress, for most materials, can be determined with a reasonable degree of precision from curves showing the range of stress up to 10 million repetitions.*

Fatigue Range at Zero Mean Stress as Compared with the Ultimate Tensile Strength. Table I shows a few typical examples of the safe range for more than 10 million cycles. It will be seen that for steels this varies from about 0·8 to 1 times the ultimate tensile strength. For certain of the modern high nickel-chromium rustless steels and for the copper alloys it is much less.

Rapid Methods of Determining Fatigue. On the end of the specimen, Fig. 11, is shown a mirror. A special optical device is used in connection with this.

Fig. 21 shows load-strain curves obtained from mild steel by this apparatus. It will be seen that at a certain range of stress, ± 15 tons, about, the first curves obtained from the specimen deviate from the straight line, and it may be inferred that these curves indicate a fatigue limit. Long time tests showed that for this and other materials the safe fatigue range did not differ very much from the value thus obtained. It was found, however, that by running cycles of stress while the specimen was being gradually loaded, the apparent fatigue range as indicated by these curves could be raised, and as a matter of fact the fatigue was actually raised by this device of gradually raising the stress while the specimen was being run. With certain plastic metals the author has shown that the apparent fatigue range can be raised more than 50 per cent by gradually increasing the load and allowing cycles to be run at each load.

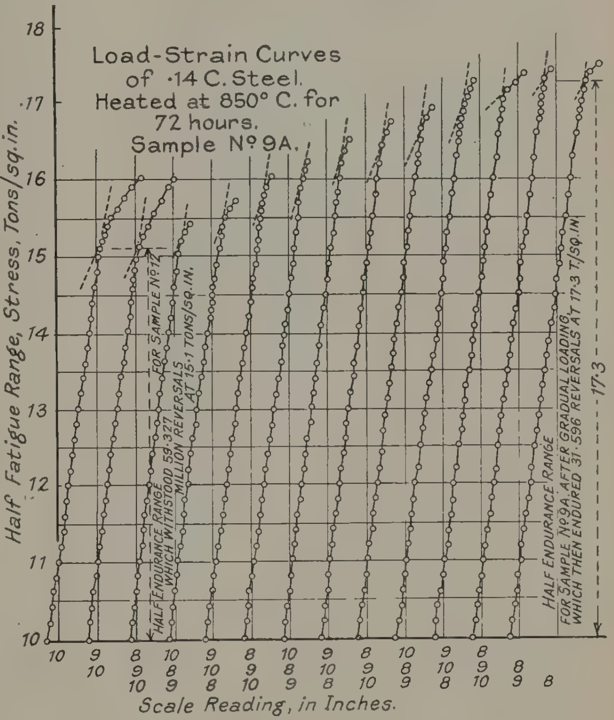

FIG. 21

Note. Piece No. 9A was repeatedly loaded to indicate 17·3 tons per sq. in. This stress was then endured by the same sample for 31·596 × 10^6 repetitions, before fracture.

A duplicate 10A was directly stressed to 17·3 tons per sq. in. but failed after 3·67921 × 10^6 repetitions

A duplicate No. 12 endured 59·327 reversals at 15·1 tons per sq. in. before breaking.

The unloading curve is not shown. The rate of loading was constant. The curves are shown only plotted above 10 tons per sq. in.

In other cases the fatigue range, as determined from such curves, differed very considerably from the fatigue range determined by plotting range of stress against numbers of repetitions. Similar torsion-strain diagrams can be obtained from the torsion machine, Fig. 13, and the other torsion machine referred to in the text. Also from the machine illustrated in Fig. 12.

The extensometer shown in Fig. 15 makes it possible to determine similar curves from tests in the Haigh machine. The method, in certain cases, gives a very good indication of the fatigue limit to be expected, but it must be checked by the long time runs and, to obtain correct results, care must be taken not to raise the limit artificially.

Fractures Under Repeated Stresses. When specimens of mild steel, and many other materials, are broken by statical loads they show very considerable ductility. In tension they elongate very considerably and show reduction of area, while in torsion they twist through, probably, several complete revolutions before fracture. When specimens fail under repeated stresses they fracture, however, almost like perfectly brittle materials. Figs. 22 and 23 show two fractures, one obtained in the laboratory, the other from a machine tool. There was no apparent change in shape or dimensions. Such fractures often show large crystal faces, and from this appearance the mistake is frequently made in assuming that the material is large grained. The structure shown by the fracture does not show the grain structure of the material, but rather the tearing away along particular planes of a large number of crystals. Only by polishing and etching can the real grain structure be seen under the microscope.

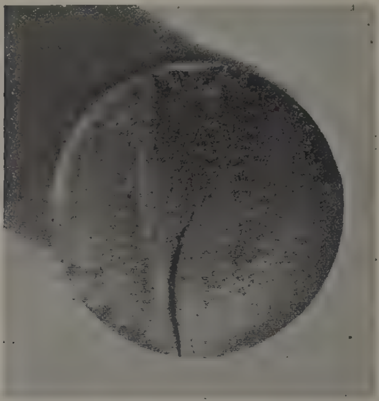

FIG. 22. CUPRO-NICKEL BROKEN UNDER REPEATED STRESSES

No apparent plasticity

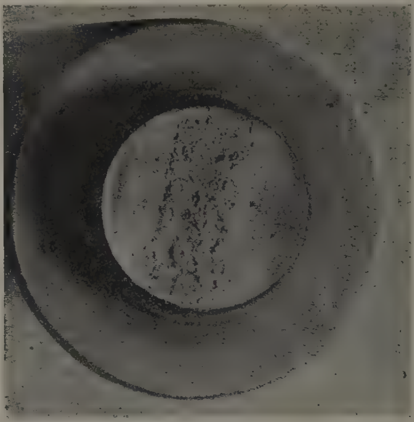

FIG. 23. STEEL BORING BAR BROKEN IN PRACTICE

No apparent plasticity

MATHEMATICS FOR ENGINEERS

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LESSON XXVIII

INTEGRATION OF SINES AND COSINES

In the last lesson we learned how to differentiate sines and cosines, and we have now to invert the formulae and obtain the corresponding ones for integration. Since $\frac{d}{d\theta}(\sin\theta) = \cos\theta$, we must have

$$\int \cos\theta \,.\, d\theta = [\sin\theta]$$

and since $\frac{d}{d\theta}(\cos\theta) = -\sin\theta$, we have

$$\int \sin\theta \,.\, d\theta = [-\cos\theta]$$

We must be careful of the algebraic signs in these two formulae; they are the opposite of those of the differentiation formulae; the integral of a sine is *minus* a cosine, and the integral of a cosine is a sine.

Another point of importance is that the angle must be thought of in radians, so that the limits of integration will be in radians; for finding the values of the sines and cosines, however, it is convenient to use degrees, since the tables are constructed for degrees. It is convenient to write the limits in both forms—radians and degrees, outside the square brackets on completing the first step in the integration, as will be shown in the examples.

Before coming to examples, however, it is convenient to generalize the formulae to include multiples of θ. By the formulae of the last lesson, with a slightly different notation, we have $\frac{d}{d\theta}(\sin n\theta) = n \cos n\theta$, i.e. on differentiating, we *multiply* by n. Integration is the inverse of differentiation, so that in integrating we must *divide* by n,

so, $$\int \cos n\theta \,.\, d\theta = \left[\frac{1}{n}\sin n\theta\right]$$

and, in a similar manner,

$$\int \sin n\theta \,.\, d\theta = \left[-\frac{1}{n}\cos n\theta\right]$$

Illustrative Examples.

$$\int_{0\cdot1745}^{0\cdot6981} \cos\theta \,.\, d\theta = \Big[\sin\theta\Big]_{0\cdot1745\,=\,10^\circ}^{0\cdot6981\,=\,40^\circ}$$
$$= (\sin 40^\circ) - (\sin 10^\circ)$$
$$= 0\cdot6428 - 0\cdot1736 = 0\cdot4692.$$

2. $$\int_0^{\frac{1}{4}\pi} \sin 3\theta \,.\, d\theta = \Big[-\tfrac{1}{3}\cos 3\theta\Big]_0^{\frac{1}{4}\pi\,=\,45^\circ}$$
$$= (-\tfrac{1}{3}\cos 135^\circ) - (-\tfrac{1}{3}\cos 0^\circ)$$
$$= +\,0\cdot2357 + 0\cdot3333 = 0\cdot5690.$$

3. To find the average value of the sine between 0° and 90°.

To find the average value, we integrate over a range thereby finding the area under the graph, and to find the average value, i.e. the average height of the graph, we divide by the base (on the graph), i.e. by the range. In our case, the range is from 0 to $\frac{1}{2}\pi$ (radians), so that the average value will be

$$\frac{1}{\frac{1}{2}\pi}\int_0^{\frac{1}{2}\pi} \sin\theta \,.\, d\theta = \left[-\frac{2}{\pi}\cos\theta\right]_0^{\frac{1}{2}\pi\,=\,90^\circ}$$
$$= \left(-\frac{2}{\pi}\cos 90^\circ\right) - \left(-\frac{2}{\pi}\cos 0^\circ\right)$$
$$= 0 + \frac{2}{\pi} = 0\cdot6366.$$

In integrating squares of sines and cosines, or products, we shall find it necessary to use some of the trigonometrical formulae given in Lessons XIX and XX. We give a few examples to show how this is done.

4. $$\int_0^{\frac{1}{3}\pi} \sin^2\theta \,.\, d\theta = \int_0^{\frac{1}{3}\pi} \tfrac{1}{2}(1 - \cos 2\theta)\, d\theta$$
$$= \Big[\tfrac{1}{2}(\theta - \tfrac{1}{2}\sin 2\theta)\Big]_0^{\frac{1}{3}\pi\,=\,60^\circ}$$
$$= \tfrac{1}{2}\left(\frac{\pi}{3} - \tfrac{1}{2}\sin 120^\circ\right) - 0$$
$$= \tfrac{1}{2}(1\cdot0472 - 0\cdot4330) = 0\cdot3071.$$

5. $$\int_0^{\frac{1}{2}\pi} \sin\theta \sin 2\theta \, d\theta$$

$$= \int_0^{\frac{1}{2}\pi} \tfrac{1}{2}(\cos\theta - \cos 3\theta)\, d\theta$$

$$= \left[\tfrac{1}{2}(\sin\theta - \tfrac{1}{3}\sin 3\theta)\right]_0^{\frac{1}{2}\pi = 90^\circ}$$

$$= \tfrac{1}{2}(\sin 90^\circ - \tfrac{1}{3}\sin 270^\circ) - (0)$$

$$= \tfrac{1}{2}(1 + \tfrac{1}{3}) = \tfrac{2}{3}$$

As a last example we will find the work done per stroke against friction in the pump of the example worked in Lesson XXVII. With the notation of that example, if f is the friction coefficient, the head lost by friction is $\frac{2flV^2}{gd}$ ft.

$$= \frac{2 \times 0{\cdot}005 \times 40 \times 21{\cdot}3^2 \sin^2 \pi t}{32{\cdot}2 \times \frac{1}{4}}$$

$$= 22{\cdot}5 \ \sin^2 \pi t \text{ ft.}$$

The weight of water delivered in a small interval, δt, is

$$\tfrac{1}{4}\pi d^2 V \times 62{\cdot}4 \,.\, \delta t$$
$$= \tfrac{1}{4}\pi \times \tfrac{1^2}{4} \times 62{\cdot}4 \times 21{\cdot}3 \sin \pi t \,.\, \delta t$$
$$= 65{\cdot}3 \sin \pi t \,.\, \delta t \text{ lb.}$$

Therefore the work done in this interval is

$$22{\cdot}5 \sin^2 \pi t \times 65{\cdot}3 \sin \pi t \,.\, \delta t$$
$$= 1470 \sin^3 \pi t \,.\, \delta t \text{ ft.-lb.}$$

Also, the time taken for one stroke is 1 sec., so we must, in order to find the work done per stroke, integrate from 0 to 1; thus

$$\text{Work/stroke} = \int_0^1 1470 \sin^3 \pi t \,.\, dt.$$

Now $\sin^3\theta = \frac{1}{4}(3\sin\theta - \sin 3\theta)$ so this integral is

$$\int_0^1 1470 \times \tfrac{1}{4}(3\sin\pi t - \sin 3\pi t)\, dt$$

$$= \left[\frac{1470}{4}\left(-\frac{3}{\pi}\cos\pi t + \frac{1}{3\pi}\cos 3\pi t\right)\right]_0^1$$

$$= \frac{1470}{12\pi}(-9\cos 180^\circ + \cos 540^\circ) -$$
$$(-9\cos 0^\circ + \cos 0^\circ)$$

$$= \frac{1470 \times 16}{12\pi} = 624 \text{ ft.-lb.}$$

EXERCISE No. 42

1. Evaluate: $\int_{0\cdot1745}^{0\cdot8727} \sin\theta \,.\, d\theta$; $\int_{0\cdot1745}^{0\cdot8727} \cos\theta \,.\, d\theta$; $\int_0^{\pi} 3\cos\frac{1}{2}\theta \,.\, d\theta$; $\int_0^{\frac{1}{2}\pi} \cos^2\theta \,.\, d\theta$; $\int_0^{\frac{1}{3}\pi} 6\cos\theta\cos 4\theta \,.\, d\theta$.

2. The acceleration of a machine part is given by the formula $6 \sin 2\pi t + \sin 6\pi t$. The velocity is zero when $t = \frac{1}{4}$, and the displacement is zero when $t = 0$. Find the formulae for the velocity and displacement.

3. Obtain by calculus the area of a segment of a circle by putting $y = r\sin\theta$ and $x = r\cos\theta$ in the area integral.

4. By dividing up the surface of a sphere into narrow zones, and showing that the area of one of them is $2\pi r^2 \sin\theta \,.\, \delta\theta$, deduce that the area of the surface is $4\pi r^2$.

FURTHER RULES IN DIFFERENTIATION

Function of a Function. In the last lesson we developed and used the formula

$$\frac{ds}{dt} = \frac{ds}{d\theta}\frac{d\theta}{dt}$$

and mentioned that it was an important and useful formula. We used it when it was possible to express s simply in terms of θ, and θ simply in terms of t, but did not know the formula for $\frac{ds}{dt}$ direct. Expressing it in terms of other letters

$$\frac{dy}{dx} = \frac{dy}{du}\frac{du}{dx}$$

we shall show how it can be used to differentiate more complicated functions of x, by the help of an intermediate variable, u.

As a first instance, we will differentiate $\sqrt{x^2 + 1}$. Here, if we had to *calculate* the value of y for any value of x, we should have to do the work in stages—find x^2, add 1 then take the square root. In fact, we cannot use the symbol $\sqrt{\ }$ until we have obtained $(x^2 + 1)$ as a *single* number. It may now be obvious that it will be a help to denote it by a *single* letter, the u of the above formula. Doing so, i.e. putting $u = x^2 + 1$ (and denoting $\sqrt{x^2 + 1}$ by y, it being then $\frac{dy}{dx}$ that we are seeking), we shall have $y = \sqrt{u} = u^{\frac{1}{2}}$. It is now quite simple to find $\frac{dy}{du}$ and $\frac{du}{dx}$, for

$$u = x^2 + 1 \qquad \text{so } \frac{du}{dx} = 2x\,;$$

$$y = u^{\frac{1}{2}} \qquad \text{so } \frac{dy}{du} = \tfrac{1}{2}u^{-\frac{1}{2}}.$$

We can now use the formula

$$\frac{dy}{dx} = \frac{dy}{du}\frac{du}{dx}$$

$$\text{giving } \frac{dy}{dx} = \tfrac{1}{2}u^{-\frac{1}{2}} \times 2x = \frac{x}{\sqrt{x^2 + 1}}$$

Similarly, if we have $y = (3x-2)^5$, we put $3x - 2 = u$, when $y = u^5$. Then $\frac{dy}{du} = 5u^4$, and $\frac{du}{dx} = 3$; consequently

$$\frac{dy}{dx} = 5u^4 \times 3 = 15\,(3x-2)^4.$$

We could have done this example by expanding y by the binomial theorem, but the method used is at once simpler and more convenient, as it puts the result in a more compact form.

Again, if $y = \sin^2\theta$ and we want $\frac{dy}{d\theta}$, we put $u = \sin\theta$, and then $y = u^2$. From these, $\frac{dy}{du} = 2u$ and $\frac{du}{d\theta} = \cos\theta$. Then $\frac{dy}{d\theta} = 2u \times \cos\theta = 2\sin\theta\cos\theta$.

We can also use this formula to differentiate $\sec\theta$. If $y = \sec\theta = \frac{1}{\cos\theta}$, put $u = \cos\theta$, when $y = \frac{1}{u} = u^{-1}$. We then have $\frac{dy}{du} = -u^{-2} = -\frac{1}{u^2}$, and $\frac{du}{d\theta} = -\sin\theta$. From these $\frac{dy}{d\theta} = \frac{dy}{du}\frac{du}{d\theta} = -\frac{1}{u^2} \times (-\sin\theta) = \frac{\sin\theta}{\cos^2\theta}$. The result may also be written $\tan\theta\sec\theta$.

Products. Another instance where further rules for differentiation are needed is when we have to differentiate a product of two or more factors, each involving the variable, and we shall give the formula which enables this to be done if we can differentiate each factor. It must be clearly understood that the differential coefficient of a product is *not* the product of the differential coefficients of the factors. Thus, if we consider x^5 as the product of x^3 and x^2, the true differential coefficient is $5x^4$; the differential coefficients of the factors are $3x^2$ and $2x$; the product of these is $6x^3$, and is *not* the differential coefficient of x^5.

To obtain the correct formula, put u for one of the factors, and v for the other, i.e. write

$$y = uv.$$

Then $y + \delta y = (u + \delta u)(v + \delta v)$

$$= uv + v\,.\,\delta u + u\,.\,\delta v + \delta u\,\delta v$$

so $\quad \delta y = v\,.\,\delta u + u\,.\,\delta v + \delta u\,.\,\delta v$

and then $\frac{\delta y}{\delta x} = v\frac{\delta u}{\delta x} + u\frac{\delta v}{\delta x} + \delta u\frac{\delta v}{\delta x}$

As $\delta x \to 0$, the last term tends to zero, so we have

$$\frac{\mathbf{dy}}{\mathbf{dx}} = \mathbf{v}\frac{\mathbf{du}}{\mathbf{dx}} + \mathbf{u}\frac{\mathbf{dv}}{\mathbf{dx}}$$

the formula sought.

As a first instance, let us see that it yields the correct result for x^5 when considered as the product of x^3 and x^2. Putting $u = x^3$ and $v = x^2$, we get $\frac{du}{dx} = 3x^2$ and $\frac{dv}{dx} = 2x$. Thus

$$\frac{dy}{dx} = x^2 \times 3x^2 + x^3 \times 2x = 5x^4,$$

which is correct.

If $y = \sin\theta\sin 2\theta$, we put $u = \sin\theta$ and $v = \sin 2\theta$, giving $\frac{du}{d\theta} = \cos\theta$ and $\frac{dv}{dx} = 2\cos 2\theta$, so

$$\begin{aligned}\frac{dy}{d\theta} &= \sin 2\theta\cos\theta + \sin\theta\,.\,2\cos 2\theta \\ &= \sin 2\theta\cos\theta + 2\sin\theta\cos 2\theta.\end{aligned}$$

Again, if $y = x\sqrt{x^2+1}$, we put $u = x$ and $v = \sqrt{x^2+1}$, giving $\frac{du}{dx} = 1$ and $\frac{dv}{dx} = \frac{x}{\sqrt{x^2+1}}$ (see above). Consequently

$$\begin{aligned}\frac{dy}{dx} &= \sqrt{x^2+1} \times 1 + x \times \frac{x}{\sqrt{x^2+1}} \\ &\ \frac{2x^2+1}{\sqrt{x^2+1}}\end{aligned}$$

Quotients. The corresponding rule for quotients can be derived from the rule for products—and, indeed, many quotients are conveniently treated as products in the same manner—by writing $y = \frac{u}{v} = uv^{-1}$. The differential coefficient of v^{-1} is, by the rule for the function of a function (compare the differentiation of $\sec\theta$ above) $-v^{-2}\frac{dv}{dx}$. Using this in the formula for products, we find

$$\begin{aligned}\frac{dy}{dx} &= v^{-1}\frac{du}{dx} + u \times \left(-v^{-2}\frac{dv}{dx}\right) \\ &= \frac{1}{v}\frac{du}{dx} - \frac{u}{v^2}\frac{dv}{dx}\end{aligned}$$

It is often convenient to use the common denominator v^2, and write the result

$$\frac{v\frac{du}{dx}-u\frac{dv}{dx}}{v^2}$$

As a first instance, we will differentiate $\frac{x}{x^2+1}$. Denoting it by y, and putting $u = x$, and $v = x^2 + 1$, we have $\frac{du}{dx} = 1$ and $\frac{dv}{dx} = 2x$. Our formula then gives

$$\frac{dy}{dx} = \frac{(x^2+1)\,.\,1 - x\,.\,2x}{(x^2+1)^2} = \frac{1-x^2}{(x^2+1)^2}$$

As a next example, we will find the differential coefficient of $\tan\theta$. We know that $\tan\theta = \frac{\sin\theta}{\cos\theta}$, so we put $u = \sin\theta$ and $v = \cos\theta$, giving $\frac{du}{d\theta}\cos\theta$ and $\frac{dv}{d\theta} = -\sin\theta$. The rule then gives

$$\frac{d}{d\theta}(\tan\theta) = \frac{\cos\theta\cos\theta - \sin\theta\,(-\sin\theta)}{\cos^2\theta}$$

$$= \frac{1}{\cos^2\theta} = \sec^2\theta$$

where we have used the fact that $\cos^2\theta + \sin^2\theta = 1$ in simplifying the numerator.

We could also find the differential coefficient of $\sec\theta$ by this method, after writing it as $\frac{1}{\cos\theta}$, putting $u = 1$ and $v = \cos\theta$. As a matter of fact, it is useful to remember the special case of the quotient formula when $u = 1$ (and consequently $\frac{du}{dx} = 0$), for differentiating reciprocals. Putting $u = 1$ and $\frac{du}{dx} = 0$ in our formula, we find

$$\frac{d}{dx}\left(\frac{1}{v}\right) = -\frac{1}{v^2}\frac{dv}{dx}$$

As a last instance, involving the more difficult cases of differentiation, we will obtain *accurate* formulae for the velocity and acceleration of a piston. The results of the last lesson, denoting the length of the crank by r, the length of the connecting rod by l, the angle turned by the crank by θ, and the angle made with the line of stroke by the connecting rod by ϕ, are

$$s = r\cos\theta + l\cos\phi - l \quad . \quad . \quad (1)$$

and

$$r\sin\theta = l\sin\phi \quad . \quad . \quad (2)$$

s being the displacement of the piston from the mid-point of its stroke. Differentiating the first of these with respect to t (time), and remembering that $\frac{d\theta}{dt} = \omega$, the angular velocity of the crank, we find

$$\frac{ds}{dt} = -\omega r\sin\theta - l\sin\phi\frac{d\phi}{dt} \quad . \quad (3)$$

Also, differentiating the second,

$$\omega r\cos\theta = l\cos\phi\frac{d\phi}{dt}$$

or

$$\frac{d\phi}{dt} = \frac{\omega r\cos\theta}{l\cos\phi} \quad . \quad . \quad . \quad (4)$$

(This formula for the angular velocity of the connecting rod is of some importance.) By using (2) above, we may express this in terms of θ, but it is more convenient to let it remain in terms of ϕ. Using this value of $\frac{d\phi}{dt}$ in (3) we get

$$\frac{ds}{dt} = -\omega r\sin\theta - l\sin\phi\frac{\omega r\cos\theta}{l\cos\phi}$$

$$= -\omega r\frac{\sin\theta\cos\phi + \cos\theta\sin\phi}{\cos\phi}$$

$$= -\omega r\sin(\theta+\phi)\sec\phi \quad . \quad (5)$$

To find the acceleration, we must differentiate this with respect to t. By the rule for functions of a function, we have

$$\frac{d}{dt}\{(\sin(\theta+\phi)\} = (\theta\cos + \phi) \times \left(\frac{d\theta}{dt} + \frac{d\phi}{dt}\right)$$

$$= \omega\cos(\theta+\phi)\left(1 + \frac{r\cos\theta}{l\cos\phi}\right)$$

using the value of $\frac{d\phi}{dt}$ from (4). Also, remembering the differential coefficient of $\sec\theta$, we have

$$\frac{d}{dt}(\sec\phi) = \sec\phi\tan\phi\frac{d\phi}{dt}$$

$$= \sec\phi\tan\phi\frac{\omega r\cos\theta}{l\cos\phi}$$

$$= \omega\frac{r\sin\phi\cos\theta}{l\cos^3\phi}$$

where the formula for $\frac{d\phi}{dt}$ has again been used.

Now using the product rule to differentiate (5), we find

$$\frac{d^2s}{dt^2} = -\omega^2 r \cos(\theta+\phi)\left(1+\frac{r\cos\theta}{l\cos\phi}\right)\sec\phi$$

$$-\omega^2 r\sin(\theta+\phi)\,\frac{r\sin\phi\cos\theta}{l\cos^3\phi}$$

which reduces to

$$-\omega^2 r\left\{\frac{\cos(\theta+\phi)}{\cos\phi}+\frac{r\cos^2\theta}{l\cos^3\phi}\right\} \quad . \quad (6)$$

It is possible to put this in terms of θ, by using the value of $\sin\phi$ from (2) and the value of $\cos\phi$ derived from it in the last lesson.

We will also find the angular acceleration of the connecting rod. Differentiating the formula for $\frac{d\phi}{dt}$ in (4), using the rule for a quotient gives

$$\frac{d^2\phi}{dt^2} = \omega \times$$

$$\frac{l\cos\phi\,.\,(-r\sin\theta)\frac{d\theta}{dt} - r\cos\theta(-l\sin\phi)\frac{d\phi}{dt}}{l^2\cos^2\phi}$$

Using the values of $\sin\phi$ and $\cos^2\phi$, after substituting the values of $\frac{d\theta}{dt}$ and $\frac{d\phi}{dt}$, this reduces to

$$-\ \omega^2\,\frac{r(l^2-r^2)\sin\theta}{l^2\cos^3\phi}$$

It will be seen that these exact formula are complicated, and the approximate ones deduced in the last lesson are accurate enough for many purposes.

EXERCISE No. 43

1. Differentiate with respect to the variable the following functions: $(2x+1)^4$; $\sqrt{x^2-1}$; $\frac{1}{x+2}$; $\cos^2\theta$; $\sin^5\theta$; $\theta\sin\theta$; $\cot\theta$; $\operatorname{cosec}\theta$; $\frac{x+3}{x+1}$; $\frac{x^2-1}{x^2+1}$; $\frac{1}{\sqrt{1-n^2\sin^2\theta}}$.

2. The illumination at the edge of a square table, due to a lamp at a height x above the centre is $\frac{Ix}{(x^2+b^2)^{1\frac{1}{2}}}$, I being the candle power of the lamp, and $2b$ the side of the square. Find x for maximum intensity.

3. A beam of rectangular section is to be cut from a log of diameter d. If b is the breadth, show that the moment of inertia is $\frac{b(d^2-b^2)^{1\frac{1}{2}}}{12}$ and find the value of b for this to be a maximum.

ANSWERS TO EXERCISE No. 41

1. $2\cos 2x$; $-3\sin 3x$; $an\cos nx$; $20\cos 5x + 40\sin 5x$; $1{\cdot}5\cos\frac{1}{2}x - 1{\cdot}25\sin\frac{1}{4}x$.

2. $\frac{ds}{dt} = -60\sin 10t + 10\sin 20t$;

   $\frac{d^2s}{dt^2} = -600\cos 10t + 200\cos 20t.$

3. 5·14, 6·28, 3·75 ft./sec.; + 55·8, − 17·6, − 55·8 ft./sec^2.
4. 78°, 18′, 6·42 ft./sec.
5. 36° 52′.
6. 23° 50′.
7. 42° 21′.
8. $P = 1{\cdot}5$, $Q = -0{\cdot}5$.

THE BALANCING OF ENGINES

By E. B. Cole, B.Sc.

LESSON III

In an engine the various masses that have reciprocating motion are continually being accelerated and retarded. To cause these accelerations a *force* must be supplied, and, at the beginning of each stroke when the masses are being accelerated from rest, part of the effort on the piston has to be utilized for this, so that the force available for transmitting to the crank is less by the amount of this *Inertia Force*, as it is called.

Towards the end of the stroke the inertia force becomes negative, i.e. the moving parts are now being retarded, and, due to the momentum they have gained, the force transmitted to the crank will be in excess of the piston effort by this inertia force.

The engine frame must, however, be subjected to a reaction equal and opposite to this accelerating force, which, due to its variations in magnitude and sign, will tend to cause unwelcome vibrations.

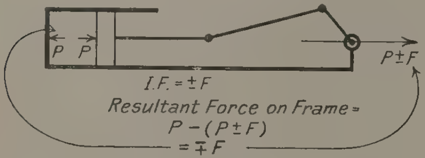

FIG. 11

It will be as well to emphasize that this reaction on the frame is equal and opposite to *the force accelerating the moving parts*, and *not* to the fluid thrust on the piston. The sketch, Fig. 11, should make this clear.

P is the fluid thrust acting on piston and cylinder head. The inertia force we will call $\pm F$, since it depends on whether the reciprocating parts are being accelerated or retarded. Then the thrust transmitted to the crankshaft and therefore from the crankshaft to engine frame $= P \pm F$ in direction shown, while the thrust on frame in the opposite direction is P. Hence resultant force on frame $= \mp F$.

With engines of more than one cylinder in one plane and parallel to one another, it will be clear that the inertia forces, which, of course, all act in the line of stroke, will cause couples on the frame tending to rotate it about an axis perpendicular to the plane of reciprocation.

It is, therefore, these forces that require to be balanced, and, in a similar manner to a revolving system, we shall refer each force to a given plane and resolve them into a single force and a couple.

We must, however, first of all determine the value of the inertia force at any instant.

Fig. 12 represents the mechanism of a reciprocating engine, the crank having turned through an angle θ *from the inner dead centre.*

Uniform angular velocity ω of the crank is assumed.

$$\left.\begin{array}{ll}\text{Length of connecting rod} & = L \\ \text{Length of crank} & = R\end{array}\right\}\text{ratio } L/R = n$$

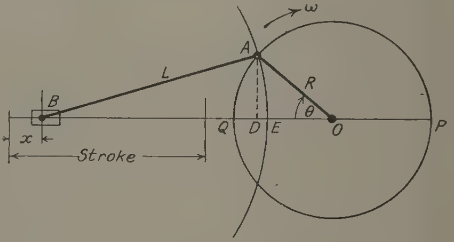

FIG. 12

With centre B and radius BA strike arc cutting OQ in E.

Draw AD perpendicular to OQ. Then the displacement of piston from the end of stroke

$$= x = QE$$

Thus
$$\begin{aligned} x &= QD + DE \\ &= [OQ - OD] + DE \\ &= [R - R\cos\theta] + DE. \end{aligned}$$

Now from Fig. 13, which shows the whole circle of centre B, we have rectangle $MD \,.\, DE = AD^2$

$$\therefore\ DE = \frac{AD^2}{MD}$$

Now, provided L is large compared with R, MD is very nearly equal to $ME = 2L$.

On this approximation therefore

$$DE = \frac{AD^2}{2L} = \frac{R^2 \sin^2\theta}{2L}$$

$$\therefore\ x = R\left[1 - \cos\theta + \frac{\sin^2\theta}{2n}\right]$$

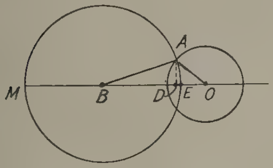

FIG. 13

The velocity of the piston will be obtained for any crank angle θ by differentiating the above—

$$\frac{dx}{dt} = R\left[\sin\theta \cdot \frac{d\theta}{dt} + \frac{2\sin\theta\cos\theta}{2n} \cdot \frac{d\theta}{dt}\right]$$

Now $d\theta/dt$ = angular velocity of crank
= constant = ω

$$\therefore\ \frac{dx}{dt} = R\omega\left[\sin\theta + \frac{\sin 2\theta}{2n}\right]$$

On again differentiating, we obtain the acceleration of the reciprocating parts—

$$\frac{d^2x}{dt^2} = R\omega\left[\cos\theta \cdot \frac{d\theta}{dt} + \frac{2\cos 2\theta}{2n} \cdot \frac{d\theta}{dt}\right]$$

$$= R\omega^2\left[\cos\theta + \frac{\cos 2\theta}{n}\right]$$

Thus, if W = weight of the reciprocating parts, the inertia force is given by

$$F = \frac{W}{g}\,\omega^2 R\left[\cos\theta + \frac{\cos 2\theta}{n}\right]$$

Now assuming that the connecting rod were *infinite*, then the piston moves with *simple harmonic motion*, since its position for any crank angle θ, Fig. 14, will be the point S in the stroke PQ, being the projection of the point A, which is moving with uniform angular velocity ω, on the diameter PQ. The acceleration of A is constant $= \omega^2 R$ directed towards O, and therefore the acceleration of the piston is the horizontal component of this, namely, $\omega^2 R \cos\theta$, always directed towards O along the line of stroke.

Thus, the first term in our expression for the inertia force, e.g. $\frac{W}{g}\,\omega^2 R\cos\theta$, represents the *simple harmonic accelerating force* required with an *infinite* connecting rod, while the second term, $\frac{W}{g}\,\omega^2 R\,\frac{\cos 2\theta}{n}$ shows the modification due to a *finite* rod.

Primary balancing, as it is called, consists in considering the simple harmonic forces assuming an infinite connecting rod, and, except in special cases of short connecting rod engines, this is sufficient for ordinary practical purposes. *Secondary* balancing takes into account the latter term of the inertia force expression.

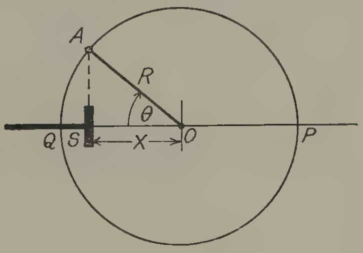

FIG. 14

Now the connecting rod has both reciprocating and rotary motion, and it can be shown that, dynamically, we produce the same effect if we consider its mass divided into two portions—one being concentrated at the crank pin and having purely *rotary* motion, the other at the crosshead or gudgeon pin and having purely reciprocating motion, this latter being included in the reciprocating masses.

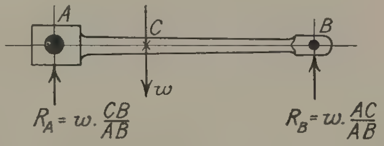

FIG. 15

It is approximately correct to take the weights of these two masses as equal to the reactions at the respective pins when the rod is supported horizontally. Thus, Fig. 15, if w = weight of rod, an amount $w \cdot \frac{CB}{AB}$ will be considered concentrated at crank pin A and will be included in the *rotating* system, while an amount $w \cdot \frac{AC}{CB}$

will be included in the *reciprocating* system, being assumed concentrated at B.

Let us now return to

Primary Balancing

The inertia force is assumed simple harmonic and is given by $\frac{W}{g}\omega^2 R \cos\theta$ for any crank angle θ being a maximum of $\pm\frac{W}{g}\omega^2 R$ at the ends of the stroke.

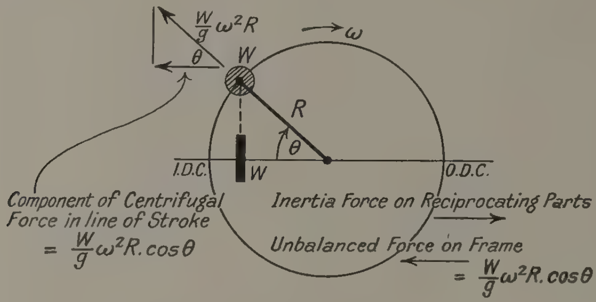
$\frac{W}{g}\omega^2 R$
ω
θ
W
R
θ
I.D.C.
W
O.D.C.
Component of Centrifugal Force in line of Stroke
$= \frac{W}{g}\omega^2 R.\cos\theta$
Inertia Force on Reciprocating Parts
Unbalanced Force on Frame
$= \frac{W}{g}\omega^2 R.\cos\theta$

Fig. 16

But $\frac{W}{g}\omega^2 R$ is equal to the centrifugal force set up by a mass of weight W revolving in a circle of radius R, so that the force on the *frame* due to the simple harmonic reciprocation of a mass of weight W is the same as *the component in the line of stroke* of the centrifugal force produced by an equal weight W revolving at crank radius R. This is illustrated in Fig. 16.

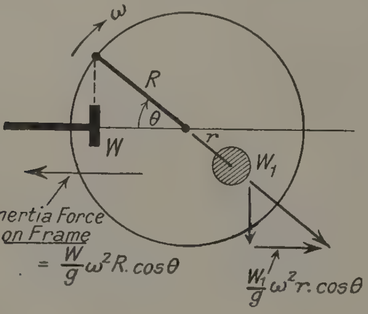
ω
R
θ
W
r
W_1
Inertia Force on Frame
$= \frac{W}{g}\omega^2 R.\cos\theta$
$\frac{W_1}{g}\omega^2 r.\cos\theta$

Fig. 17

Thus to find the unbalanced force and couple due to a system of reciprocating masses in one plane we may consider the masses as concentrated at crank radius and treat as an *imaginary* revolving system.

The closures of the polygons give us the unbalanced force and couple due to these imaginary centrifugal forces, and these closures therefore represent the *maximum values* of the unbalanced inertia force and couple acting on engine frame in regard to the given reference plane, *the actual values at any instant being their components in the line of stroke.*

Consider now a reciprocating system in which a *revolving* mass of weight W_1 is placed at radius r on the opposite side of the shaft to the crank pin, such that $W_1 r = -WR$ as indicated in Fig. 17.

This mass actually sets up a centrifugal force whose component in the line of stroke will balance the inertia force on the frame, due to the reciprocation of W, and there are no disturbances *in the line of stroke*, but an equal disturbance will now occur in a direction perpendicular to the line of stroke, due to the other component of the centrifugal force of W_1.

Thus a rotating mass arranged in this manner cannot completely balance a reciprocating mass, but can only change the plane in which the force acts.

To effect a balance, we can consider the masses as an imaginary revolving system and determine the magnitudes and directions of two extra masses at crank radius to effect a balance as a *revolving* system, as previously explained. These masses if *reciprocated* will then give balance to the reciprocating system.

Such additional masses are termed "bob-weights," but it will be realized that each bob-weight requires an extra crank and connecting rod to operate it which may cause excessive production costs.

With revolving masses, which are usually in given fixed positions, the addition of two extra masses involves no complications, but with a reciprocating system, this multiplicity of cranks is a decided disadvantage; however, it will be seen that if there are a sufficient number of cylinders, the reciprocating masses of two may be arranged to act as balance weights for the remainder by suitably arranging the crank positions.

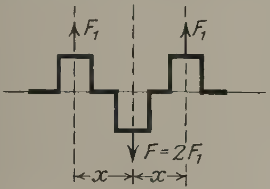

FIG. 18

Thus the problem with a reciprocating system is to arrange the crank positions so that balance is effected without the necessity of adding extra masses.

This is done on large triple and quadruple expansion high speed marine engines, where vibrational considerations are of paramount importance.

It should be noted, however, that while arranging the crank positions will effect balance, the further problems of arranging the cranks so that the turning moment may be fairly even, and so that the engine may be readily started from any position, must be considered.

For instance, Fig. 18 represents a three-crank engine, the cranks all being in one plane, the middle one carrying twice the reciprocating mass of either of the two outside cranks and at 180° to them. This system is in complete primary balance, but the turning moment will be most irregular and difficulty of starting from some positions would be experienced.

Actually for marine work it is usual to employ four cranks—two L.P. or two I.P. cylinders being provided for triple expansion engines—since by suitable arrangement of the cranks and adjustment of their masses complete primary balance can be obtained, while easy starting from any position and an even turning moment can also be secured.

Let us take two examples—

1. In a four-cylinder two-stroke Diesel engine the reciprocating masses of each cylinder weigh 600 lb., and the cranks are arranged at 90°, the order in end elevation being 1, 3, 2, 4. The pitch of the cylinders is 2·5 ft., stroke 18 in., and the revolutions per minute 250.

Determine the maximum values of the unbalanced primary force and couple in regard to a central reference plane, and also state their instantaneous value when crank (1) has turned through 90° from the inner dead centre.

Fig. 19 shows the crank arrangement.

Remembering that for balancing purposes we can consider the reciprocating masses as concentrated at their respective crank pins, and forming an imaginary revolving system we see at once that there is *no unbalanced force*, the forces balancing one another since the masses are equal and the cranks opposed to one another.

The *couples* will be proportional to the distance of the cranks from the reference plane, since the masses are equal.

Calling, for convenience, the pitch of the cylinders $= 2a$, and the right-hand side of the reference plane negative, then the couples are proportional to

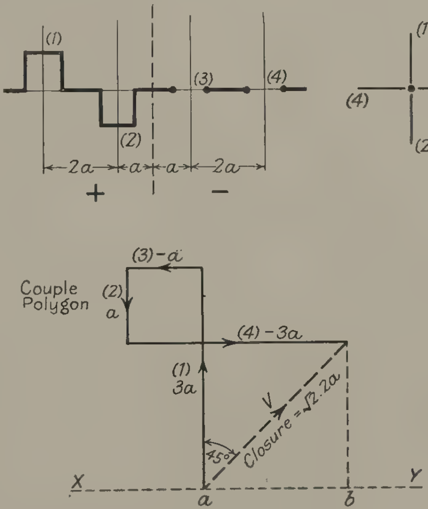

FIG. 19

1 . .	$3a$
2 . .	a
3 . .	$-a$
4 . .	$-3a$

The couple polygon is then shown in Fig. 19, the couples having been taken in clockwise crank order.

The closure represents, in magnitude and position relative to the cranks, the resultant imaginary centrifugal couple acting. Actually this gives the maximum value of the unbalanced inertia couple acting in the plane of reciprocation tending to turn the engine about a central axis perpendicular to this plane. This will occur when the cranks have turned into such a position that this vector V lies in the line of stroke XY. At the instant depicted crank (1) is at right angles to XY, and the instantaneous value of the couple is the component of V in the line of stroke, i.e. ab.

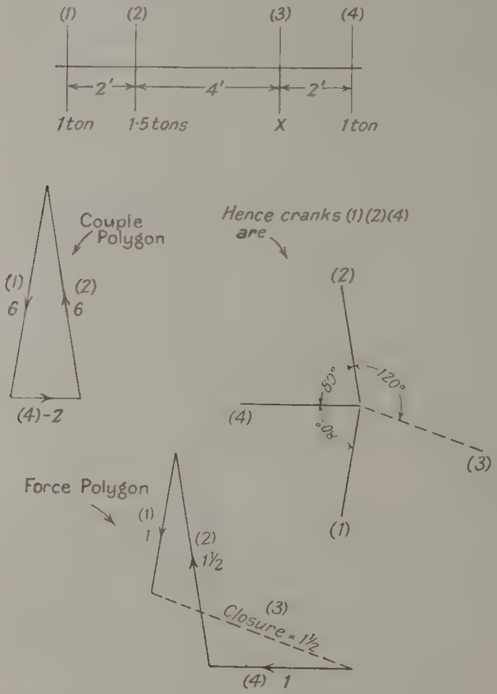

FIG. 20

Thus from couple polygon

$$\text{Maximum value} = \sqrt{2} \cdot 2a \cdot \frac{W}{g}\,\omega^2 r$$

$$= \sqrt{2} \cdot 2{\cdot}5 \cdot \frac{600}{32} \cdot \left(\frac{2\pi \cdot 250}{60}\right) \cdot \frac{9}{12}$$

$$= \underline{34{,}100 \text{ lb.-ft.}}$$

Instantaneous value at given instant

$$= 34{,}100 \cos 45^\circ$$

$$= \underline{24{,}100 \text{ lb.-ft.}}$$

EXAMPLE 2. The cylinder pitches of a four-crank engine are shown in Fig. 20, and the reciprocating masses of cylinders (1), (2), and (4) are 1, 1½, and 1 ton respectively. Determine the crank positions, and the reciprocating mass for cylinder (3) for complete primary balance.

Take a reference plane at the unknown mass, and draw up a table as for a revolving system. Since the masses are all considered concentrated at crank radius, this radius may be taken as unity for the purpose of drawing the vectors.

Plane	Distance from Ref. Plane x	$\omega^2 r$ = unity	
		Imaginary Centrifugal Force $\propto W$	Imaginary Centrifugal Couple $\propto Wx$
1	6 ft.	1	6
2	4	1·5	6
$\boxed{3}$	0		0
4	− 2	1	− 2

All the couples are known in *magnitude*, and since the couple polygon is to be a closed figure and, in this case, is a triangle, it can at once be drawn, the sides indicating the relative positions of cranks (1), (2), and (4). Fig. 20. Note direction of crank (4).

The force polygon consists of four sides, three of which are now known in magnitude and direction, so that the closure will give the magnitude of the unknown mass (3) and the direction of its crank. The magnitude of this mass is 1·5 tons, which might have been deduced from the symmetry of the engine.

(CONCLUSION)

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XXVII

Development of Design. We have remarked that machine construction consists, in a great measure, of building with existing details and, while this is the general case, one ought to guard against the habit of assuming the detail part of a design to be stereotyped. It is true that development in detail design is slow for the good reason that engineers hesitate to alter what has proved reliable. One may be forgiven for thinking that the proverb, " the more it changes the more it remains the same," has a particular application to machine design. This fact does, at times, tend to discourage the young designer and turn his initiative into a stoic copying, but such an attitude is neither good nor justifiable.

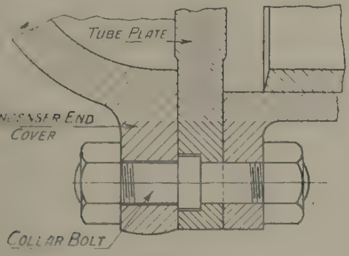

Fig. 1

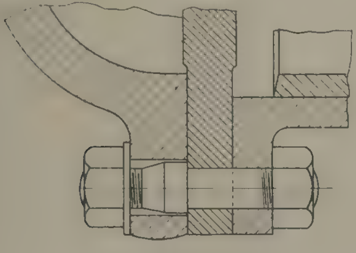

Fig. 2

One may examine a score details with the object of improving the same and find only one suitable for change, yet the one improvement will be worth the effort.

We may now turn our attention to a few elementary cases in which the designer's alertness for development would result in a gradual improvement of the detail.

Development for Economy. An example is given in Fig. 1, which shows an old type collar bolt for a condenser tube plate joint. The machining involved in making the recess in the tube plate to suit the bolt is not merited by the advantage of providing the larger facing for the bolt nut. Consequently, the design was modified to the type shown in Fig. 2, the addition of the washer compensating for the reduced area of facing on the door in way of the nut. This development is one which would occur to one using Fig. 1 as a guidance. Having recognized the improvement it requires only the courage of one's conviction to make the change.

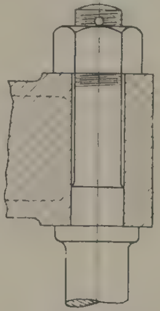

Fig. 3

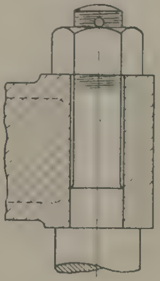

Fig. 4

A further example is given in the pump column shown in Fig. 3, the maximum diameter of this detail is such that it could be most economically made from bar. This would entail reducing the section at the ends and in the body, but it will be observed that the reduction of the body section is for appearance only. In the interest of economy we may modify the detail as shown in Fig. 4.

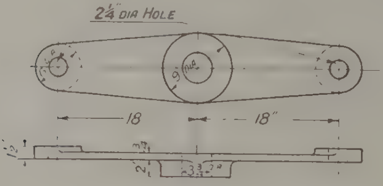

FIG. 5

A final example of development in design for economy of manufacture is given in Fig. 5, which shows a reciprocating engine pump lever designed to be made from a solid steel forging. For the dimensions given such a detail would be costly, and the process of forging would prove the most costly part of its manufacture. The elimination of forging from the manufacture would result in a considerable saving, and this economy may be obtained by building the part with a steel plate, pads, and small forged steel flange as shown in Fig. 6.

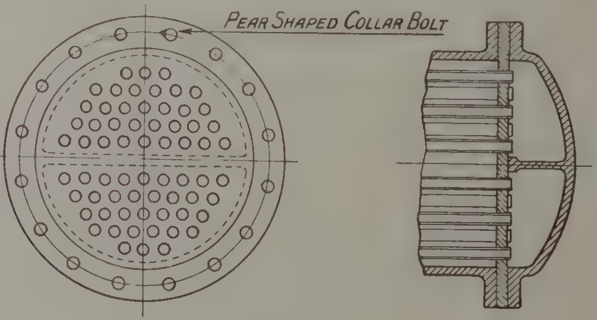

FIG. 7

Development from Experience. Mishap, and not foresight alone, may play a part in the improvement of detail design. It may happen that a detail will serve its purpose quite effectively for a number of machines and then fail through a common misfortune, the misfortune being, in this case, responsible for the improvement.

Such a case is given in Fig. 7, which shows a cover for a drain cooler. It will be evident that, for correct service in dividing the tubes into two flows, the cover must be fitted in a certain position relative to the body. With all the cover bolts similar, correct alignment of the cover is dependent upon the care of the fitter. Such a design is unnecessarily dangerous and may be made quite safe by simply making one bolt of special form, say, a pear shape, as shown in dotted lines.

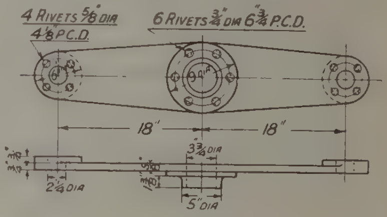

FIG. 6

A further example of development in design following experience is given in Fig. 8, which shows a device for preventing two valves being closed at the same time. Such a requirement occurs when a combined valve chest controls the exhaust from a steam-engine to one of two directions, it being essential that one valve should always be open.

It will be seen that the locking pin *L* must be placed comparatively close to the valve *A* before valve *B* can be closed, and that this close proximity of the locking pin to the valve spindle can only obtain when the valve *A* is open. If an operator carelessly attempts to close valve *B* before opening valve *A*, the sleeve on spindle *B* will find an obstruction in the locking pin *L*. This obstruction will remind him that valve *A* must be first opened

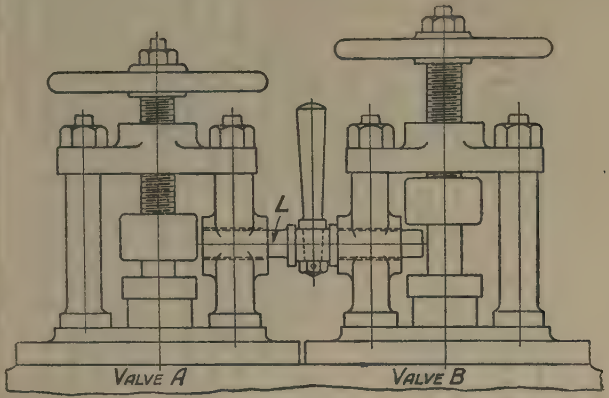

Fig. 8

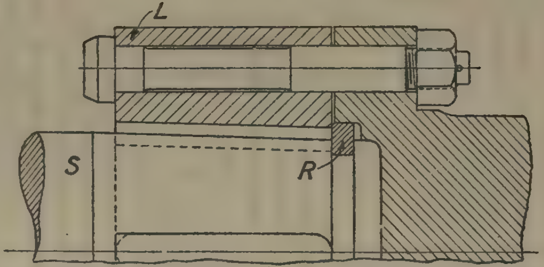

Fig. 10

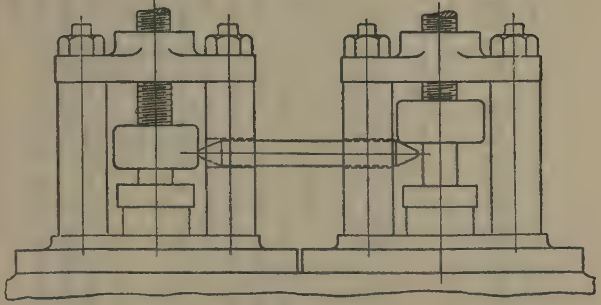

Fig. 9

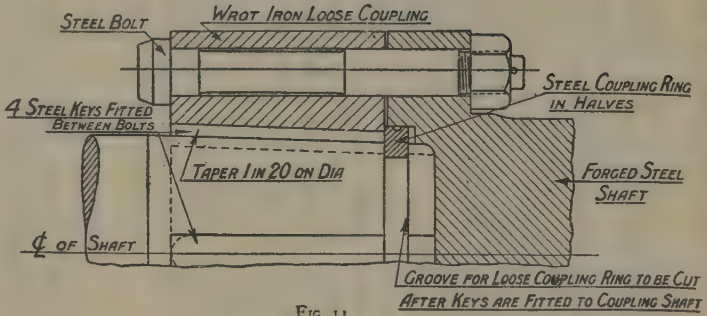
Steel Bolt
Wrot Iron Loose Coupling
4 Steel Keys Fitted Between Bolts
Steel Coupling Ring in Halves
Taper 1 in 20 on Dia
Forged Steel Shaft
℄ of Shaft
Groove for Loose Coupling Ring to be Cut After Keys are Fitted to Coupling Shaft

Fig. 11

Details of Valve Locking Device and Shaft Loose Coupling

and so the device makes the valve chest *fool-proof*.

We may imagine the circumstance in which an operator, ignorant of the steam system, endeavouring to close valve *B* (with valve *A* in the closed position) and straining the locking pin. Such a strain may produce a malalignment sufficiently large to prevent the required movement of the pin *L*.

Such an experience would suggest an improvement to the design as shown in Fig. 9, where the movement of the locking pin *L* becomes automatic. Further, it will be observed that the modified design is cheaper to produce.

Development to Assist Manufacture. It may happen that a design is quite good with regard to strength and reliability, but poor when considered relative to the ease of manufacture. This concern applies generally to those machine parts made in such small numbers that special experience of making the part has not had time to grow. In such cases it is advisable to aim at a design in which the machining and fitting shall be as simple as possible and not demand any special thought or consideration.

Such an example, where the required number is small and the care required in fitting equally great is given in Fig. 10, which shows a loose coupling fitted to a ship's propeller shaft to enable the shaft to be withdrawn from the ship through the stern tube.

In considering the method of fitting the loose coupling *L* to the shaft *S* it will be evident that great difficulty may be experienced in fitting the coupling tight on both the shaft and keys, at the same time keeping the coupling face in line with the ring *R*. Much of this difficulty may be avoided if the key-ways are cut to the shaft end, as shown in Fig. 11. This will allow the loose coupling to be fitted tight on the shaft and the keys to be driven home, binding the sides rather than the top of the coupling key-way. Further, the note suggesting the groove for the coupling ring to be cut after the keys are fitted will allow this part of the alignment to be accomplished quite easily. Finally, a groove added to the outside of the ring will enable the bottom half of same to be easily held in position while engaging the ring to the corresponding recess in the solid coupling.

EXERCISE

Choose a number of machine details from available drawings and engineering periodicals and, applying the above method, essay a development in design in as many cases as possible.

OUTLINES OF LOCOMOTIVE ENGINEERING

By A. Morton Bell, O.B.E., M.I.Mech.E., M.I.Loco.E.

LESSON IX

THE TENDER

In addition to primarily being a fuel and water storage for the locomotive, the tender forms a useful and important adjunct. Originally, a box on wheels for the wood fuel, surmounted with a barrel for feed water, it has developed into a compact, if heavy, and carefully designed addition to the locomotive structure.

The illustration, Fig. 37, shows one possessing some unique features, supplemental to the fact that it carries all the fuel and water required, it is also of considerable service when utilized as a retarding element in controlling heavy trains on steep grades.

The tank accommodates some 6,000 gal. of water, and the bunker (which is designed to be self-trimming) 14 tons of coal. To facilitate taking water, the whole side of the tender is made available for receiving the end of the water column, thus obviating the necessity to stop an engine within a foot or so of any exact spot. The outline adopted also permits of the engine-men having a good view of the track, if running tender first, or whilst engaged in shunting operations.

To reduce weight, tenders of this description have been built with welded joints, thus excluding riveting and rendering unnecessary many of the butt strips, angles, etc., required for it.

The weight of the tender shown in running order with coal and water is 73 tons.

The capacity and gross weight of this example are considerable, but it is necessary to carry large quantities of coal and water if heavy trains are to be hauled over long distances without stops, or delays, for locomotive purposes. On more favoured routes with shorter runs, smaller tenders are employed, and in the case of the 4–6–0 express engines of the G.W.R. (Fig. 3), a very compact and medium-sized tender is provided. It is fitted, however, with a water-pick-up arrangement, to enable fresh supplies of water being taken whilst running; a suitably made scoop is lowered into a long, narrow water tank placed between the tracks on a level section of the line, and the water is impelled up the pipe into the tank by the rapid motion of the engine. It is not always possible to secure the last requirement—a level section of line of sufficient length—and, further, there are difficulties where there are unfenced railways in sparsely populated countries, from wild animals straying on the lines to drink.

As already noted, considerable assistance is expected from a locomotive tender in controlling trains with its brakes; the brake gear should be well designed of an equalizing type, to ensure all brake blocks pressing on the wheels in unison, and with similar power.

The brake rigging, as it is termed, consists of a number of cross-beams suspended from hangers, attached to the tender framing. The cross beams are connected up with the pull-rod of the brake operating mechanism, and are also attached to the brake blocks acting against the wheels. By introducing equalizing levers among the rigging, all blocks can be arranged to bear with equal pressure on the wheels when power is applied.

A form of tender which has met with favourable adoption in America and abroad consists of a large, cylindrical tank for the water, and a sloping, self-trimming bunker for the coal, next to the locomotive; a mechanical apparatus operated by air is often placed herein to push the coal forward as required for firing.

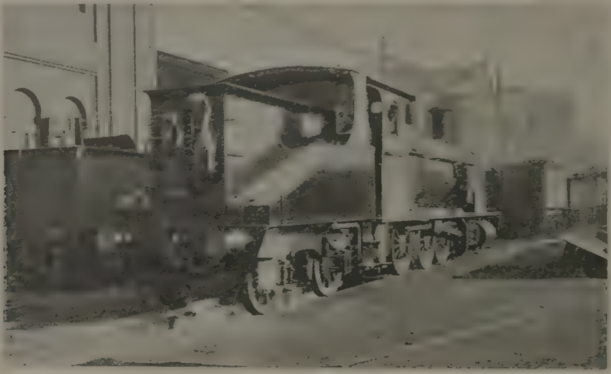

FIG. 37. TENDER, NEW SOUTH WALES GOVERNMENT RAILWAYS

It is recommended that the tool boxes, cupboards for men's belongings, sanding gear, etc., be conveniently arranged at the locomotive, or foot-plate, end of the tender. An approved indicator, or gauge, to show the water level in the water tank is now usually fitted.

The coupling between the engine and tender is a detail on which some ingenuity has been bestowed, as it is most essential that whilst a tight and reliable attachment is made, there should be ample flexibility to allow of easy passage over track cross-overs, etc. A recent design provides a stout, central, articulated draw bar, with a ball and socket joint at each end. This takes care of both draft and buffing stresses; slotted safety links are provided on either side. In other designs, short, or dummy, buffers are fitted, to ensure a smooth running connection,

GAS ENGINES AND PRODUCERS

By Arnold Rimmer, B.Eng.

LESSON XII

POWER GASES

Consideration having now been given to the construction and working of gas engines, attention will be turned to the subject of the various gases on which they may be operated, and the methods of manufacturing certain classes of the same.

Theoretically, any gas that is capable of burning in the presence of oxygen with an evolution of heat in the process, may be utilized in a gas engine. In practice, various considerations, especially that of cleanliness, are involved, and the gases used contain hydrogen or carbon (or both) as the combustible constituent.

In order to understand the processes involved, it is necessary to consider a few chemical reactions, in which the following elements are included—

Element	Chemical Symbol	Atomic Weight
Hydrogen	H	1
Carbon	C	12
Nitrogen	N	14
Oxygen	O	16

It should be noted that hydrogen, nitrogen, and oxygen are *di-atomic*, i.e. a molecule consists of two atoms and is, therefore, represented by H_2, N_2, and O_2 respectively. Carbon, on the other hand, is *mono-atomic*, as the molecule consists of a single atom.

Hydrogen is odourless and non-poisonous, and burns with a non-luminous flame, producing water vapour according to the equation

$$2H_2 + O_2 = 2H_2O$$

The heat evolved is about 290 B.Th.U.'s per cub. ft., though if the water vapour is condensed the additional "latent heat" causes the *higher calorific value* to be about 340 B.Th.U.'s per cub. ft.

Carbon monoxide (CO) is produced when carbon is burnt in the presence of a limited supply of oxygen

$$2C + O_2 = 2CO$$

If the combustion is complete, however, carbon dioxide (CO_2) will be formed, and this may be reduced to carbon monoxide by being passed over a further portion of the heated carbon

$$C + CO_2 = 2CO.$$

This, of course, is the action which takes place in an ordinary fire of fair depth. The blue flame which is to be seen at the top of the fire is the carbon monoxide burning to form once again carbon dioxide

$$2CO + O_2 = 2CO_2$$

When used in connection with gas engines, of course, this gas is not allowed to burn in this manner until it reaches the engine cylinder. Its "higher" calorific value is about the same as that of hydrogen, but it is really superior to the latter, from the point of view of heat evolved, since there is no water vapour in the products of combustion. It should be noted that in the actual gas engine cylinder the temperatures prevailing are such as to prevent the condensation of water vapour and, consequently, the latent heat from the same is not available. Carbon monoxide is odourless but very poisonous, and adequate ventilation is essential in all buildings in which plants for the manufacture of this gas are situated.

Nitrogen and oxygen, of course, are the chief constituents of the atmosphere, which is a mixture (not a chemical compound) of these two gases (together with traces of other gases) in the following proportions—

	Oxygen	Nitrogen
By weight .	23%	77%
By volume .	21%	79%

Oxygen does not burn, but is a strong supporter of combustion. Nitrogen and also carbon dioxide do neither, and so they are simply diluents when present in power gases.

There are a large number of compounds (solids, liquids, and gases) of hydrogen and carbon to which the general name of "hydrocarbons" is applied. The two most generally found in power gases are—

(*a*) Marsh gas or methane (CH_4) having a lower calorific value of 955 B.Th.U.'s per cub. ft., and burning with a non-luminous flame to form carbon dioxide and water vapour

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

(*b*) Olefiant gas or ethylene (C_2H_4) which has a still larger calorific value (1,588 B.Th.U's. per cub. ft.) and forms the same products of combustion though it burns with a highly luminous flame

$$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$$

Power Gases. The gases already considered are present in varying proportions in the following "power gases," which are the chief ones used for gas engine operation—

1. **Town Gas.** This is ordinary lighting gas, produced by the distillation of soft coal in closed retorts, the gas being afterwards carefully cleaned. (The residue in the retorts, of course, is coke.) This gas, consisting chiefly of carburetted hydrogen, has a high calorific value but is also costly, and its use is generally restricted to small engines. It has the advantage of being available for immediate use from the supply mains.

2. **Producer Gas.** In the manufacture of *water gas*, steam is blown through a mass of incandescent carbon, being thereby decomposed and hydrogen and carbon monoxide formed. This process absorbs heat, and it is necessary to stop it at regular intervals and blow air only through the carbon again to raise its temperature. The gas produced during the latter process is usually termed *producer gas* (in contradistinction to *water gas*), and may be collected separately and utilized or else blown to waste.

In connection with gas engine operation, however, the term "producer gas" is generally applied to that formed when a mixture of air and steam is passed through coal or other fuel which is burning in a special furnace. The gas needs careful cleaning, especially to ensure the removal of all tar, etc. The combustible constituents are carbon monoxide, hydrogen, and small quantities of gaseous hydrocarbons. The calorific value is much lower than that of town gas but, as a smaller proportion of air is required, the heating value of the mixture is not greatly altered. It is much cheaper than town gas, and a producer of some form is used in the majority of gas engine installations to-day (Fig. 69).

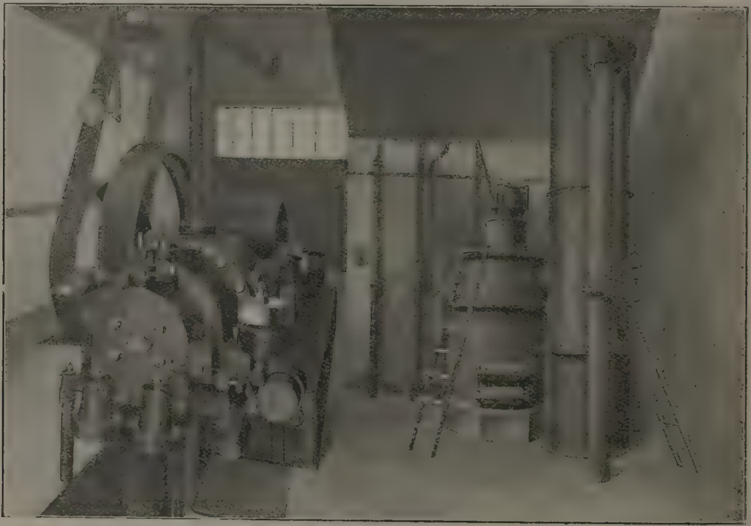

FIG. 69. ENGINE AND PRODUCER GAS PLANT (NATIONAL)

3. **Blast Furnace Gas.** As previously stated, this is a "waste product" obtained during the reduction of iron ores, etc. It is of low calorific value and needs careful cleaning, as it contains fine attritive dust which would, of course, be detrimental to the working parts of a gas engine. Since coke is the usual fuel used in blast furnaces

there is no tar in the gas, and carbon monoxide is the chief combustible. The great merit of this gas is that of economy, since it can be made available with but little expenditure and would be otherwise wasted.

4. **Coke Oven Gas.** This is produced during the preparation, from specially selected coals, of a hard pure coke for use in blast furnaces. Hydrogen and methane are the chief constituents, and the calorific value is almost equal to that of town gas.

5. **Natural Gas.** The decomposition of vegetable matter under a restricted supply of oxygen causes the generation of natural gas in the earth itself. It consists chiefly of methane and is found mainly in America and Russia, where it is used extensively for gas engine operation.

A few further particulars of these gases are given in the table below. The calorific values stated are average figures—

Gas	Lower Calorific Value (per cub.ft.)	Ratio of Air to Gas for Complete Combustion	Lower Calorific Value of Mixture (per cub.ft.)
	B.Th.U.		B.Th.U.
Town gas	550	5·25 : 1	88·0
Producer gas	150	1·1 : 1	71·5
Blast furnace gas	100	0·75 : 1	57·0
Coke oven gas	500	5 : 1	83·5
Natural gas	900	9 : 1	90·0

Natural gas is not available to any extent in this country; lighting gas is expensive and the use of blast furnace and coke oven gas is limited, as a rule, to the works producing the same. Producer gas, on the other hand, has played a very important part in the general development of the gas engine, and has now a wide application. It is proposed to confine attention here to this class of power gas, together with the consideration of the construction and operation of the various forms of gas plant in which it is made.

Producer Fuels. At the present time there are a great number of fuels which may be used in a gas producer, among which are those shown in the table in the next column.

In addition, recent developments have shown the possibility of using much "waste matter," such as sawdust, spent tan, coffee and cocoa husks, cotton seeds, sugar cane refuse, etc.

	Fuel	Remarks
(*a*)	Anthracite coal	Nearly all carbon and contains very little volatile matter.
(*b*)	Bituminous coal	Contains a greater percentage of volatile matter and consists of 70–80 per cent carbon.
(*c*)	Coke	The residue from the coal used for gas making. It is about 90 per cent carbon, the rest being chiefly ash.
(*d*)	Charcoal	Obtained when the volatile constituents are driven off wood.
(*e*)	Peat	Decayed vegetable matter found in "bogs."

Types of Gas Plants. The heat evolved by the burning of the above fuels is considerable and, in order to prevent the formation of clinker, etc., and also reduce radiation losses, it is necessary to control the temperature attained in the producer. This is usually effected by mixing steam with the air passing through the fire. A portion of the heat is absorbed in "dissociating" the steam and, in addition, the gas is enriched by the hydrogen which is formed

$$C + H_2O = CO + H_2$$

Gas producers may be divided into two main classes, viz.,

1. Suction plants.
2. Pressure plants.

In the latter, steam is generated in a separate boiler, and the mixture of air and steam forced through the generator by means of a blower or steam injector. In suction plants, however, the air and steam are caused to flow through the incandescent bed of fuel by the rarefaction of pressure produced by the engine during the induction stroke, the steam being formed in a "vaporizer" through which the air passes, an intimate mixture of the two being thus ensured. An average fuel consumption for suction gas plants is 1 lb. of anthracite (or 1¼ lb. of coke) per b.h.p. hour for continuous running. The water required for steam making and for cleaning the gas is usually about 1½ gal. per b.h.p. hour.

In addition to firms specializing on gas plant construction, several gas engine manufacturers now make their own producers, etc. Variations in the same are chiefly in matters of detail and, in the next Lesson, a brief description of a few representative plants will be given.

DIESEL ENGINES

CONSTRUCTION, OPERATION, AND MAINTENANCE

By A. Orton, A.M.I.Mech.E.

LESSON XII

MAINTENANCE—(*contd.*)

The Fuel Pump. The packing of the operating spindles and the plungers (if packings be used) needs to be kept oil-tight, and very occasionally the suction and delivery valves require regrinding on their seats; beyond this there is little to go wrong.

One of the troubles associated with the working of the fuel pump is "air-locking," that is, the accumulation of small volumes of air in the working chamber or in the space below the delivery valve. This air, by being compressed and re-expanded during the delivery and suction strokes respectively, effectively prevents the full quantity of oil being delivered. When this trouble occurs, as it sometimes does after an engine has been standing idle for some time or after a dismantling of the pump or piping, it usually requires the opening up of the delivery valves so that the fuel oil can be pumped or allowed to flow up past them and thus to permit the escape of the air. Once the spaces below the delivery valve are clear the trouble will cease, as the plunger then works against solid oil and a full charge must go forward during the delivery stroke.

In the solid injection system with pump injection this point is of extreme importance, for even a minute quantity of air in any part of the system subject to the intermittent pump pressure will seriously interfere with the injection, if it does not stop it altogether. For this reason, in all types of pumps care is taken in the design of the pump and piping system to avoid pockets or recesses in which air might lodge.

When an engine has stood for any length of time it is necessary before starting up to prime the delivery piping to the fuel valves. Means of doing this are always provided, either by operation of the main plungers themselves or by small auxiliary plungers fitted in the delivery chamber of the pump. This is done to ensure full pipes when the engine starts, so that immediately the fuel valve comes into action oil is delivered into the cylinder. When priming in this way, it is advisable to disconnect the pipe from the fuel valve casing and allow the oil, when the pipe is full, to flow out at the end rather than into the casing. Too much oil in the casing would cause a very heavy charge to enter the cylinder, which might lead to excessive pressure and violent bumping when the engine begins to fire. A small by-pass valve is often provided in the delivery piping near the fuel valve casing to allow this priming operation to be more conveniently carried out.

It is the correct adjustment of the fuel pump that ensures equal distribution of the work done by the respective cylinders of a multi-cylinder engine. Separate adjustment is always provided for each cylinder, usually by means of a tappet screw under the suction valves of the pump in a normal design, where there is a separate plunger for each individual cylinder. This is a matter that may require attention in running, even though the settings may have been correct previously. For instance, if one plunger becomes more leaky than another and there was no means of correcting this otherwise, the extra loss could be allowed for by giving that particular plunger a little longer effective stroke. Differences in the tightness of the fuel pump valves might be allowed for in a similar way, but, of course, the proper thing to do in that case would be to put the valves right.

Air Compressor, Intercooler, and Air Receivers. In the early days of the Diesel engine there was often a considerable amount of trouble with the air compressor. The range of pressures and temperatures in the respective stages of compression was high, the valves were heavy and difficult to maintain in good working order and often the method of lubrication was unsuitable. Modern designs, however, are much improved, such as by better proportioning of the respective stages of compression, thereby equalizing the ranges of pressure and temperature, the use of three, or on large engines even four, stages of compression instead of two, simpler and better design of the valves, the old mitre-seated type being now almost universally displaced by valves of the plain disc type, absolute control of cylinder lubrication by the use of mechanical sight feed lubricators and the use of more suitable lubricating oils. Hence, the compressor may now be regarded as quite as reliable as the main parts of the engine, and it is exceedingly rare in these days that any serious trouble occurs.

Practically only the valves require any special attention. As already mentioned, these are generally of the simple disc type and automatic in action. Occasionally they require to be dismantled and cleaned as they tend in time to become sticky with carbonized lubricating oil. Careful attention to the cylinder lubrication and restriction of the amount of oil used to the minimum necessary for efficient working are the most important factors in reducing this trouble. Too much lubricating oil is sure to lead to carbonization of the valves and also to similar trouble with the compressor pistons.

The intercoolers have no working parts, but still demand a certain amount of regular attention while the engine is running, owing to the unavoidable accumulation of water that takes place in them. A certain amount of oil from the compressor is also carried into them, which, if allowed to collect in sufficient quantity, may form with the water and under the action of the high pressure air a slightly acid emulsion. The presence of this water is due to the moisture contained in the air which is taken into the compressor and which is condensed in the intercoolers. The quantity of it varies, therefore, according to the state of the atmosphere, and may amount to quite a considerable quantity on a damp or foggy day when the air is heavily laden with moisture. The amount of oil that is carried over depends chiefly upon the efficiency and correct adjustment of the compressor lubrication.

Regular drainage of this liquid is, therefore, a necessity, and provision is always made for this by including in the design suitable settlement spaces in which drain valves are fitted.

Intercooler coils, which are usually made of copper, also suffer from a liability to corrode and erode under the action of the liquid already referred to in conjunction with the scouring action of the hot, high pressure air. This applies particularly to the coils of the high pressure stage. Care must, therefore, be taken to avoid failure of the coils from this cause, and the now recognized method is to inspect the coils at stated intervals and to keep a regular check on their weight. If corrosion is taking place the weight will gradually diminish and, when that reduction has reached some agreed amount, say, 25 per cent, the coil is replaced by a new one. This is an important matter, as the bursting of an intercooler coil causes a temporary stoppage of the engine and may result in the fracture of the surrounding water jacket, unless this is provided with suitable relief valves.

The starting air and blast air receivers, like the intercoolers, accumulate a certain amount of water condensed from the atmospheric air which is pumped into them and this requires to be drained off, suitable cocks being always provided for this purpose. If this is not attended to in the case of the blast air receiver, it is possible for some of this water, which, as already explained, may contain a certain amount of lubricating oil, to be carried with the blast air into the fuel injection valve casings and, by mixing there with the fuel oil to cause misfiring or irregular running of the engine. Cases have occurred in the past where the dirty carbonized oil from the compressor has been carried in this way right into the fuel valve casing, and there has caused choking of the pulverizer. Such a case, of course, points to defective methods of lubrication of the compressor and could scarcely occur in a modern design with the better control of the supply of lubricant. The important point in this connection is to recognize that in all compressed air systems the accumulation of water cannot be avoided, and that provision must be made for, and attention regularly given to, its removal.

General Maintenance. We cannot here refer in detail to all those matters of ordinary procedure which are necessary to maintain a Diesel, or any other type of internal combustion engine, in an efficient state of working, that is to say, the rectification or renewal of parts subject to wear, such as the main and connecting rod bearings, cylinder liners, pistons, piston rings, exhaust valves, etc., the correction or readjustment of valve settings, the maintenance and correct alignment of crankshafts (a matter of considerable importance in multi-cylinder engines), the choice of the most suitable lubricants and their correct use, and possible treatment after use, the proper regulation of the water circulation and the keeping of the engine in a clean and efficient state. In such matters the Diesel engine, like other engines, must receive its due measure of attention and care, if satisfactory and continued service is to be expected. When such attention is given, then it may be taken that there is scarcely any limit to the life

(*Continued on page* 1429)

SOLDERING, BRAZING, AND WELDING

By E. Arthur Atkins, M.I.Mech.E.

LESSON IV

POINTS TO BE NOTED IN ARC WELDING

One of the unfortunate things in connection with fusion welding is that the process appears to be so simple that an ordinary mechanic may imagine that he can carry out the operation with little or no instruction. But in actual practice it is found that the best type of welder is one who is capable of thinking out ways and means of doing his work in the best possible manner.

In the past fusion methods of welding came into bad repute very largely through inexperienced and unthinking operators. There are all kinds of points which it is necessary for a welder to attend to if he is to obtain the best results. A few of these are as follows—

Generator. It should not be forgotten that the generator, whether it is driven by motor, steam, or petrol engine, requires regular attention, and if this be given the operator will be well repaid in the higher efficiency of the machine.

Switchboard, etc. The connections on the switchboard, cable joints, and electrode holder should be carefully made so that there will be no leakage of current. Care should also be taken that all parts of the cables are properly insulated so as to avoid the danger of a "short" which may result in a skin burn or fire.

Skin Protection. A welder should always wear gloves and see that all parts of the body are covered so as to keep the injurious rays of the arc from striking directly upon the skin. Neglect of this precaution may cause the skin to become inflamed and very painful. Screens which are used for the protection of the eyes should also be sufficiently large to cover the face.

A good sized leather apron should be worn not only to protect the clothing from being burned by showers of sparks, but also to offer additional resistance to the passage of the injurious rays to the skin.

Celluloid collars should not be worn when in the vicinity of any kind of welding operation as collars of this description may readily be set on fire by flying particles of molten metal or oxide. It is necessary that the neck should be protected, but by a material of non-inflammable character.

Eye protection. It is a good plan to make a rule never to look at the arc with the eyes unprotected, particularly is this true when striking the arc. As good eyesight is of the greatest value to the welder, no precaution should be neglected that will enable him to keep his eyes in the best possible condition.

Goggles. Care should be taken to use only goggles that are suitable to the eyesight of the particular operator, and for this purpose it is worth while experimenting to find out which tint of glasses will enable him to do his work with the greatest ease and the least strain to the eyes. Glasses which are too clear may impair the eyesight; on the other hand, if they are too dense eye strain may be set up with resulting inflammation.

A simple method of testing the suitability of glasses is to weld for a short time, stop, and then look in a direction away from the work, and if a white or yellow spot is retained in the vision it is an indication that the glasses are not dense enough. If, however, the welder has difficulty in seeing the work about the arc then the glasses are too dense.

After the most suitable glasses have been obtained it is a wise provision to have a clear glass lens fixed on the outside of the coloured glass in the frame so as to protect the coloured glass from being destroyed through the splashing of metal or oxide on same.

It is also advisable to have cardboard side protectors as shown in Fig. 13 attached to the spectacles, as these will prevent flashes of light striking from the sides.

Polarity of Current. Usually it is advisable to have the positive terminal attached to the work, and the negative to the electrode, as this method of connection gives a larger quantity of heat at the work end of the arc where it is required on account of the mass of metal being very much greater than the electrode. With some systems of welding or classes of work the experienced operator may find it necessary to reverse the polarity, but a little experimenting will soon show which polarity gives the best results.

With the metallic arc if the positive cable be attached to the holder, and the negative to the work, the arc is usually found to be erratic in working, and it is not so easy to control the flow of the deposited metal.

If there is any difficulty in knowing which are the respective poles this can very simply be overcome by the use of a piece of pole-finding paper.

Regulation of Current. The current should be regulated so as to give a metal deposit which will not only be solid but will also be completely fused to the metal of the work, and at the same time be deposited at the maximum speed.

For long seam plate work the speed of deposit is, of course, important, whereas in odds and ends of repair work the rate of throwing down the metal is not of such importance.

If more than one run is to be made on a joint it is a good plan to have the current slightly on the high side to commence with, then, as heat is generated in the work, to reduce the current.

Sometimes it is a good method to make a first run with, say, a No. 10 gauge electrode, and a second run with a No. 8 gauge, and if a third run is required with a No. 6 gauge wire, and the current should be increased with the increasing diameter of electrode. The flow of the deposited metal will determine the increase or decrease of the current, and it is only by careful, steady practice that this can be accurately ascertained. The speed of travel along a welded seam must, of course, be related to the flow of the deposited metal.

Movement of Electrode. The travelling movement of the electrode and method of gripping it is important. A circular or a half circular motion (Fig. 14) should be adopted to ensure that both sides of the vee of the joint will be thoroughly united, but on no account must too much metal be deposited at one time or the ultimate result will be either defects from contraction or the forming of blow-holes caused by the gases generated through the oxidizing of some of the metal constituents being enclosed in the metal when it solidifies.

A circular motion of small backward and forward swing is perhaps as good a method as can be used. The metal will then deposit in a manner after the form of a solid spiral spring, and will give a very good grain on solidification.

Sometimes the operator will find it difficult to keep a steady arc, due to the vibration of a long electrode, particularly so with the lighter gauges of wire. To overcome this it is a good plan to crank or bend the electrode slightly, as this has the effect of damping the vibration of the wire, and certainly results in a better deposition of the metal. Another method which can sometimes be applied is to bend the gripped end of the electrode to a right angle (Fig. 14), and when the wire is cranked, as previously explained, this will allow the operator to commence work with the electrode at an angle of 90° to the job, and with a slight forward movement any angle can be obtained to give the best condition for depositing the metal.

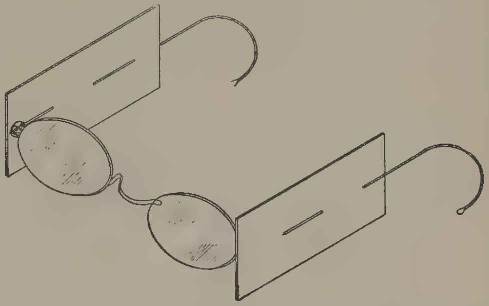

FIG. 13.

PLAIN GOGGLES WITH SIDE PROTECTION

Which must always be used when hammering the weld and if welding near a wall that is whitewashed. Side protections are made from pieces of cardboard

Fig. 14 shows in detail what has been mentioned above with regard to the holding and movement of the electrode. The arrows indicate the direction of travel, and the diagrams 1-5 the variations in movement of the tip of the electrode whilst welding and depositing the metal. No. 1 shows the half circular movement which by some operators is considered to be the best one to work to, as there is no traversing the ground a second time, but a continual sideways-forward movement.

No. 2 shows a circular or spiral movement. This is a useful method to adopt when the puddling of metal is required, care being taken that the spaces in between the spirals are closed up, and continuous contact made with the molten metal.

No. 3 is a movement which consists of going across to the right, forward a little, then to the left and forward again, and is similar to travelling along three sides of a square. This method will be found efficient in stopping the arc from attacking the surface of the walls in a drastic manner. In these movements the arc must be as short as possible.

No. 4 may be described as an inclined backward and forward movement, whilst No. 5 is just a regular zigzag forward movement. This will slightly increase and decrease the size of gap, tending to increase and reduce the temperature respectively. In some cases in practice it will be found to give advantages, but on no account should this method be applied if using heavily coated rods, or the weld will be defective in the way of slag inclusions.

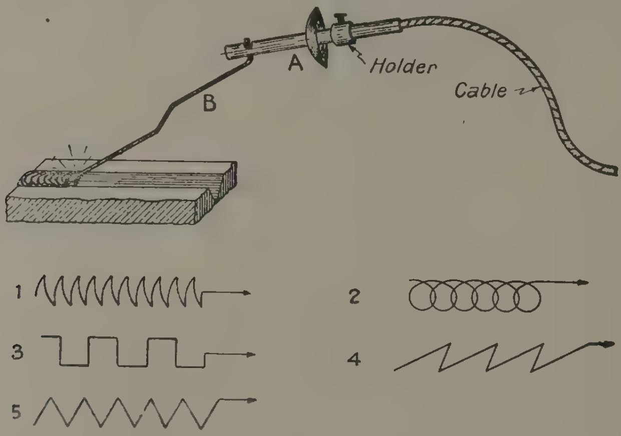

Fig. 14

A = Electrode Holder
B = Electrode with Right Angle End, also Cranked in Middle

1. Half circular travel.
2. Circular travel.
3. Three-sides of a square movement.
4. Inclined backward and forward travel.
5. Zigzag travel.

At the end of a run, or when a wire has to be replaced, the arc should always be broken as quickly as possible. If this is not carried out a black cavity will be formed in the weld caused by the gap enlarging and through increased temperature, the metal becoming burnt. When this happens the defective surface should be chipped out before being refilled with new metal. If the arc is broken quickly there is not sufficient time for an increased temperature to be set up, and consequently there will be no defective portion in the weld.

Striking the Arc. It will require some practice before one unaccustomed to welding is able to strike and maintain a steady arc, but will it be of some assistance to consider the following points—

As the pulsation of the body considerably affects the keeping of a steady arc, the operator should take care to arrange himself in as comfortable a position as possible, and where convenient be seated. In this manner he will find it much easier to damp down any vibrations that may be set up by himself.

As the weight of the cables also has a tendency to interfere with the maintaining of a steady arc these should be supported across the knee, allowing only sufficient unsupported length for easy movement of the electrode. Another plan sometimes adopted is to support the cable with the hand that holds the welding screen, with just sufficient free length to allow the holder to move freely downwards as the wire is consumed.

The elbows should be kept close to the sides, and the operator should learn to manipulate the electrode holder with a movement from the wrist. If work is carried on with the elbows held out from the body, there is a tendency for movement to take place from the shoulder with a resulting unsteady arc. As any form of stiffness militates against the obtaining of a steady arc it is as well to arrange for the cables to be as flexible as possible.

The Flashing of the Arc. As previously mentioned, care should be taken to protect the eyes from side flashes. If, however, the operator should unfortunately receive a flash from the arc on the eyes, measures should be taken to remove its effects. It is usually several hours before the full results are felt from a bad flash, and it takes about twenty hours before it passes away. It is, therefore, advisable to give a hint or two as to how to deal with this occurrence.

An eye bath, together with some boracic powder can be procured from a chemist for a few coppers. About a quarter of a teaspoonful of the boracic powder should be dissolved in hot water, and when this has cooled down to about the temperature of the body it should be poured through a piece of lint or clean linen into the eye bath, when it will be ready for use. If the eyes are bathed with this it will be found to relieve the severe inflammation considerably. Before retiring for the night a few drops of pure castor oil should be applied to the eyes, this will further relieve the pain and also act as a lubricant. Sometimes the effect will be felt during the night, but the welder should not be alarmed as this will pass away. If the pain is severe relief can be obtained by laying a cloth which has been dipped in cold water over the top of the eyes.

Many operators suffer with their eyes when first commencing to work at electric welding, but this passes away later when the eyes get used to the welding light, but it should be remembered that it is impossible for the eyes to get accustomed to the lighting of the arc without the screen, hence again the caution to take care of the eyes should be emphasized.

After a heavy day's welding an operator will find very beneficial results from having a walk in the country, or a cycle run beside green fields, the green of the grass and the trees having a very soothing effect on the eyes, in addition to the breathing of the pure air after the vitiated atmosphere that usually obtains about welding work.

Hammering of Welds. It will readily be understood that when a weld has been made by the aid of the electric arc the deposited metal through having been in a fused condition practically has all the characteristics of metal in the cast state. Therefore, to bring back some degree of toughness it is always advisable, wherever possible, in the welding of wrought iron and mild steel to hammer the deposited metal after it has become solid at a high temperature. In some cases, in addition, it is always a great advantage to anneal the joint as this will give increased toughness.

Copper as a Protecting Metal. Copper bar, plate, and sheet can be considerably utilized in connection with welding, as it is easy to bend to the shape of joint on any broken portion to which it can be attached, and easily separated after the weld has been made. Thus, for instance, if a cast-iron article is broken at some awkward part a piece of plate copper can be bent to the desired shape so as to support the broken portion. This portion can then be veed and clamped to the copper. The arc can penetrate through the vee and plate upon the copper without making an impression upon the latter. When the weld is completed the copper plate can be removed, leaving the surface of the cast-iron article quite flush, and no dressing will be required.

Copper can also be used to protect holes from being filled up when a fracture passes through same (Fig. 15), and is being repaired. A bush can be made of about $\frac{1}{8}$ in. plate copper to the diameter of the hole. This can be firmly inserted, and after the welding has taken place be readily removed.

In the welding of thin pieces of iron or steel, sheet copper can be used to cover the underside of the joint which will prevent the metal from running through. Copper can also be used when holes are being filled up, as a piece of this metal can be fixed for the bottom of a hole and the metal deposited on same.

In filling in holes the end of the electrode should be continually pointed to the walls of the hole, at the same time the travel taking place round and round. If the metal begins to run too freely this, of course, is an indication of too much heat, and the current should be reduced.

Welding Cast Iron. One of the commonest complaints about electric welds which have been made on castings is that the material after welding is so difficult either to machine or to cut, and also that the edge of the weld is so extremely hard and brittle, that the least vibration in thin castings will cause a fracture alongside the weld. This is due to the high temperature to which the walls of the weld are raised, the graphite in the cast iron completely combining or going into solution with the iron and forming an extremely hard compound known as cementite or carbide of iron. The quick rate of cooling, due to the heavy mass of metal conducting the heat from the small quantity of metal which is molten along the weld line, causes the latter to be "chilled," that is to say, that there is not sufficient time for the graphite which is in solution to be thrown out again in cooling. There are two obvious remedies, one is to reheat the weld and to allow for very slow cooling away from all draughts of cold air. The other remedy is to use such a flux on the electrode as will oppose the carbon going into solution. This latter method is one which possesses all the elements that make for the successful welding of cast iron, and from experiments that have been carried out there seems now to be no doubt that this will become the common way by which cast iron in the future will be welded to give a soft deposit.

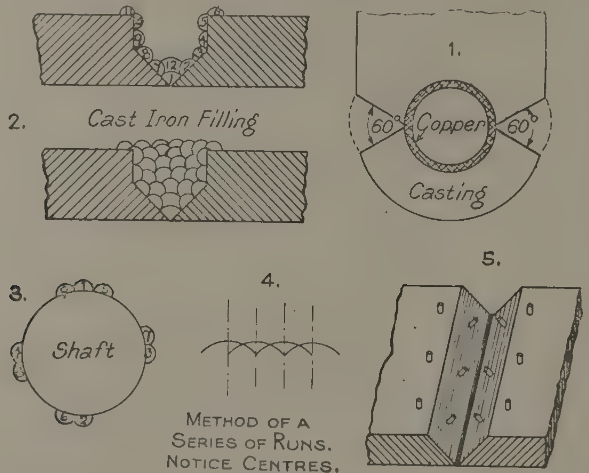

FIG. 15.

1. Method of protecting hole with copper.
2. Method and order of runs on joint of cast iron.
3. Method and order on building up shaft.
4. Method of depositing metal for an overlay pad.
5. Method of pegging cast iron.

In acetylene welding it is necessary to use a ferro-silicon rod, but with electric arc welding the conditions are entirely different, and a mild steel wire electrode with a suitable flux will give the correct quality of deposit if the work is done properly.

The greatest care must be exercised in the welding of cast iron if the best result is to be obtained in the use of these specially coated electrodes. If it can be conveniently done the work should be pre-heated to a temperature somewhat higher than the melting point of lead, and this can be quite easily tested by drawing a piece of sheet lead along the surface and observing if the remaining streak is melted. Care must be taken to adjust the current correctly so that the electrode will run freely but not too rapidly. When the arc is struck and proper contact made with the molten metal the point of the electrode must be moved in such a manner as to more or less puddle the molten metal, and this is particularly so with regard to the edges of the weld. By this puddling the flux material will be worked into the molten metal, and as previously mentioned offer resistance to all the graphite going into solution.

After the weld is completed it must be protected in such a manner as to ensure slow cooling, both from the point of view of avoiding contraction fractures, and also to leave the deposited metal in a soft condition fit for machining.

There are many additional hints that might be given in connection with the welding of cast iron, but perhaps the following two or three will be sufficient—

In welding together two flat surfaces the tendency on cooling is for the joint to sink and the plate to settle in the form of a gutter. Therefore, to avoid this it is a good plan to pack up the joint a little out of the flat, so that when cooling has taken place the plate may be drawn back to the level. All kinds of methods are sometimes adopted, by bolting, clamping, etc., to restrain the contraction of the cast iron after cooling, but these, it need hardly be said, are of very little avail, as the force of contraction in cast iron is so enormous and the metal usually so brittle that the casting will break either before or after being released. The careful avoidance of contraction strains is one of the things that the good welder is always thinking about on cast-iron jobs, and this, in many cases, can be overcome by one or other method.

On some jobs it is a good plan, if no machining is required afterwards, to set up an opposing stress in the metal by the application of mechanical force, either by wedging or some other way, but this can only be applied to a limited number of jobs.

Wherever possible it is always advisable to reinforce the metal when repairing a cast-iron fracture, the reinforcing being done by building up as much as convenient at that part of the joint where there would be no objection to the extra metal put on to give additional strength. The extra reinforcement, of course, takes the part of a pad on the top of the deposited metal in the vee, and it is not a bad plan to make this pad about half as wide again as the top of the vee.

Motor cylinders should always be pre-heated if they are to be electrically welded, as very serious contraction strains will be set up in the metal when cooling, if the cylinder is not carefully heated before welding commences.

The inexperienced may easily be led astray by the ease with which a crack in the cylinder appears to be repaired without pre-heating, but he will be quickly disillusioned when he finds an ugly crack appear when the cylinder has cooled down. Sometimes the fracture may be so fine in the weld that it is hardly perceptible.

Pegging of Cast Iron. When jobs have to be welded in graphitic cast iron, as mentioned above, it is best to peg the joint as shown in Fig. 15, No. 5, by drilling holes, tapping, and screwing in a series of mild steel pegs. The pegs will be merged and welded into the deposited metal, give strength to the joint, and thus enable the operator to make the best of a bad job.

The above form of spongy cast iron can generally be tested by a hammer and chisel, and if it is found to break away in powder this will give an indication of the difficulties that a welder is up against in attempting to make a success of this kind of metal.

Finally, it can be taken as a general rule that without a foundation of good metal it is extremely difficult to make a presentable job, and often a waste of time and good material to attempt same.

(Conclusion)

DIESEL ENGINES

(*Continued from page* 1423)

of a well-designed Diesel engine, for, by the renewal from time to time of the worn parts, its efficiency can quite easily be continuously maintained equal to that of a new engine.

The above remarks on maintenance apply, of course, more particularly to the air injection type of engine, but in a general way to the solid injection type also. There are naturally with that type special precautions to take and points to watch in connection with the fuel injection system, such as the maintenance of thoroughly tight joints to withstand the high oil pressures, the cleaning and care of the fuel pump valves and injection nozzles, avoidance of air-locking and so on. These, however, are points on which it is not proposed to enlarge here and, in any case, they are mostly of a very practical nature, quickly recognized and understood by any competent man who may be called upon to deal with them.

STEAM TURBINES

By W. J. Kearton, M.Eng., A.M.I.Mech.E., A.M.Inst.N.A.

LESSON XI

THE LJUNGSTRÖM TURBINE

A very successful double-motion radial-flow impulse-reaction steam turbine is that developed by the two Swedish engineers, Messrs. Birger and Fredrik Ljungström, about the year 1913. The then new machine was a radical departure in steam turbine construction, and embodied several ingenious devices to reduce blade tip leakage, dummy leakage, gland leakage, and conduction and radiation losses. The machine is constructed in this country by the Brush Electrical Engineering Co., Ltd.

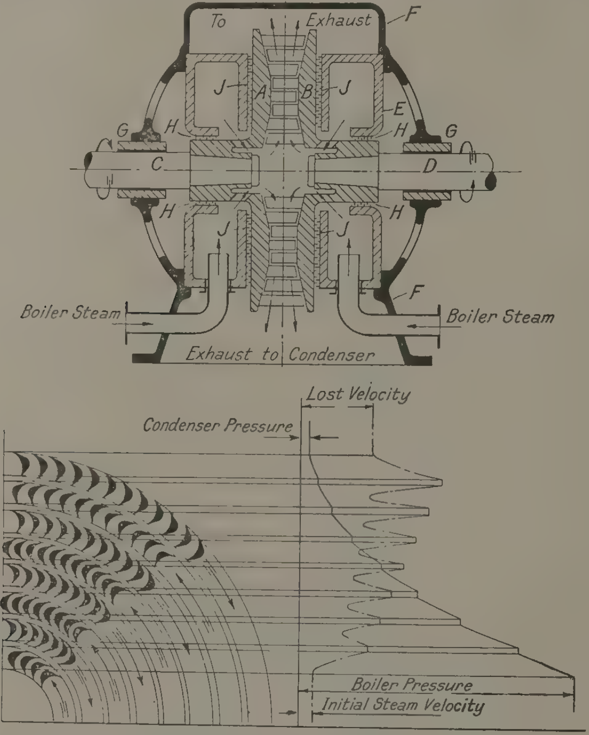

Fig. 45. Diagrammatic Arrangement of Double-motion Radial-flow Reaction Turbine

(Ljungström Type)

Note. In each ring of blades, the blades extend around the complete circumference

Fig. 45 shows a diagrammatic arrangement of the turbine. Discs *A* and *B* are attached to the shafts *C* and *D* respectively, and rotate in opposite directions. *E* is the steam chest, and *F* the exhaust casing to which the bearings *GG* are suitably attached. Leakage of steam from the steam chest *E* is minimized by suitable shaft glands *H*.

The axial thrust on the discs *A* and *B*, due to steam pressure and tending to force the discs apart, is balanced by a gradually falling steam pressure from chest to exhaust casing in the "dummy" labyrinth *J*.

Steam is supplied by two pipes leading from the side of the exhaust casing into the steam chest *E*. It passes from thence through a number of holes in the hub of the rotor into the centre of the blade system, through which it flows radially outward in all directions.

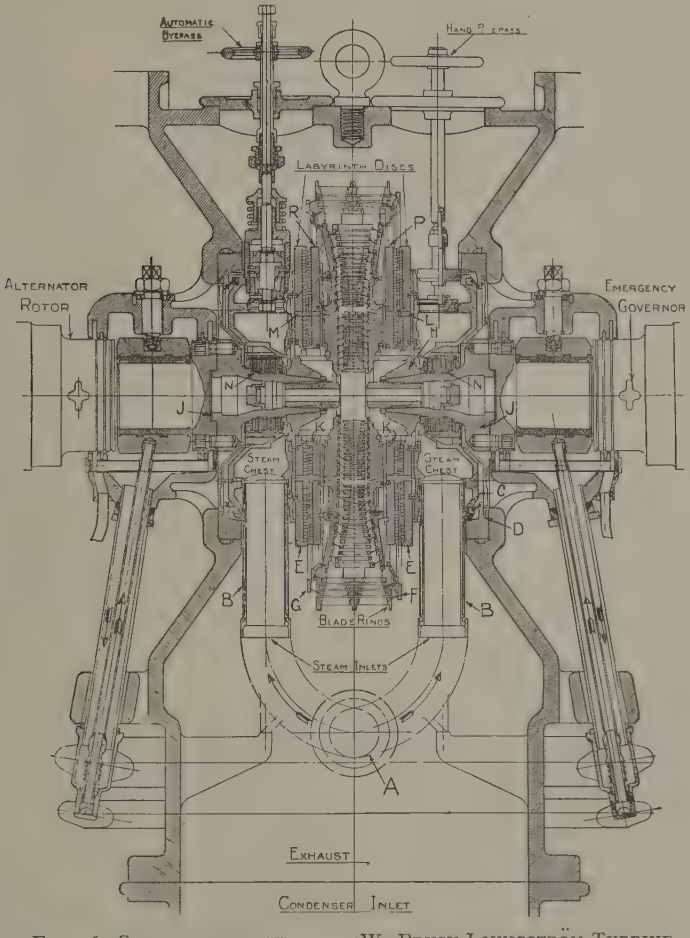

FIG. 46. SECTION THROUGH 1,500 KW. BRUSH-LJUNGSTRÖM TURBINE

A section through a 1,500 kW. Brush-Ljungström turbine is shown in Fig. 46. *F* and *G* are the two rotor discs, the hubs *H* of which are bored to a sharp taper and attached to the stub shafts *J*. Each of the latter is bolted to the shaft of an alternator. The alternators run in opposite directions and are operated in parallel. An exciter is driven from one of the alternators only, but serves both machines, of course. Each rotor disc is equipped with a number (twenty in this instance) of concentric rings of reaction blades, so arranged that when the rotors are placed together coaxially, the blade rings of one rotor fit in between the blade rings of the other.

The steam chest is a steel casting *C*, which is secured to a seating ring *D* by *bulb expansion rings*. The latter are such an important feature of the turbine that a few words will be devoted to them. The rings are of steel and have a cross-section shaped somewhat like a dumb-bell. Each side of the ring is gripped in an annular recess in the parts to be attached, the recess being closed as shown by the diagram, Fig. 47b. Provided the parts are properly designed—even though they may be rotating at a high speed—the only strains on the ring are (1) that due to the weight of the part supported; (2) that caused by differential expansion, due either to temperature variation or mechanical strain; and (3) the torque on the blade ring. Owing to the yielding nature of the ring, it exerts comparatively little constraint on the parts in the event of differential expansion. A further advantage, exemplified in the attachment of the steam chest C to the seating ring D, is the small amount of heat conduction consequent upon the small area of the ring.

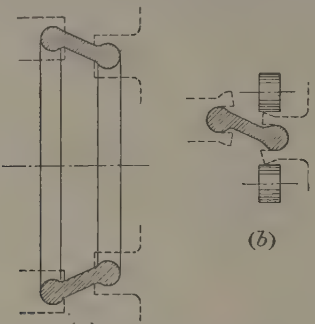
(a)
(b)

Fig. 47. Bulb Expansion Ring

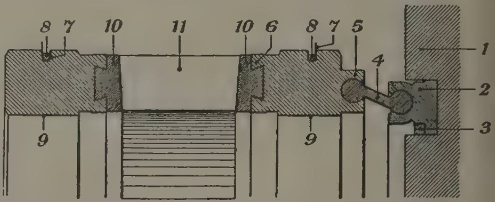

Fig. 49. Section through Ljungström Turbine Blade Ring

The fixed labyrinth discs E are attached to the chest C, and the rotating labyrinth discs K

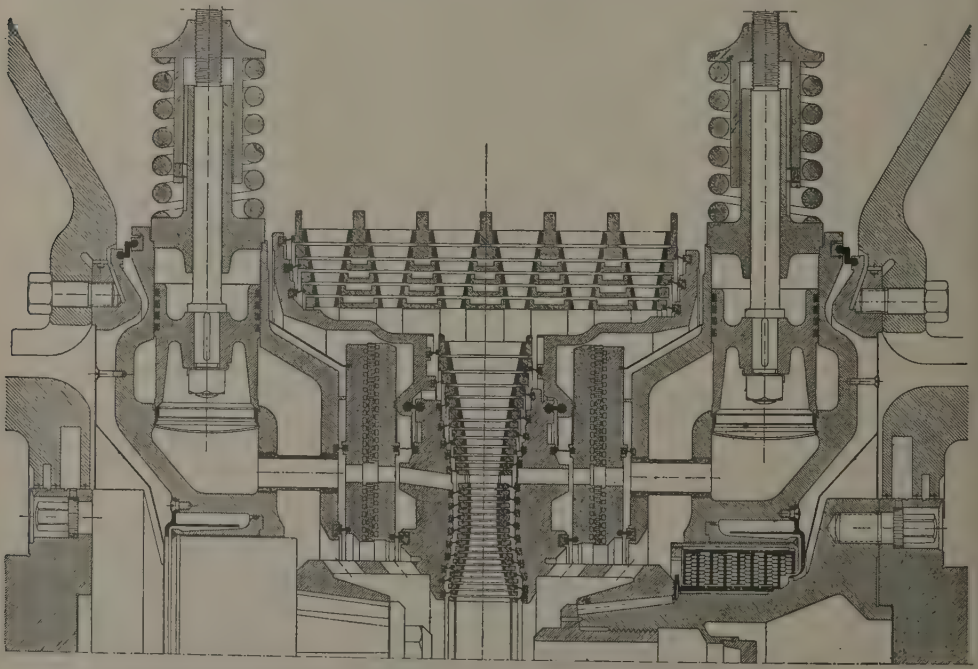

Fig. 48. Section through the Upper Half of a 5,000 kW. Ljungström Turbine

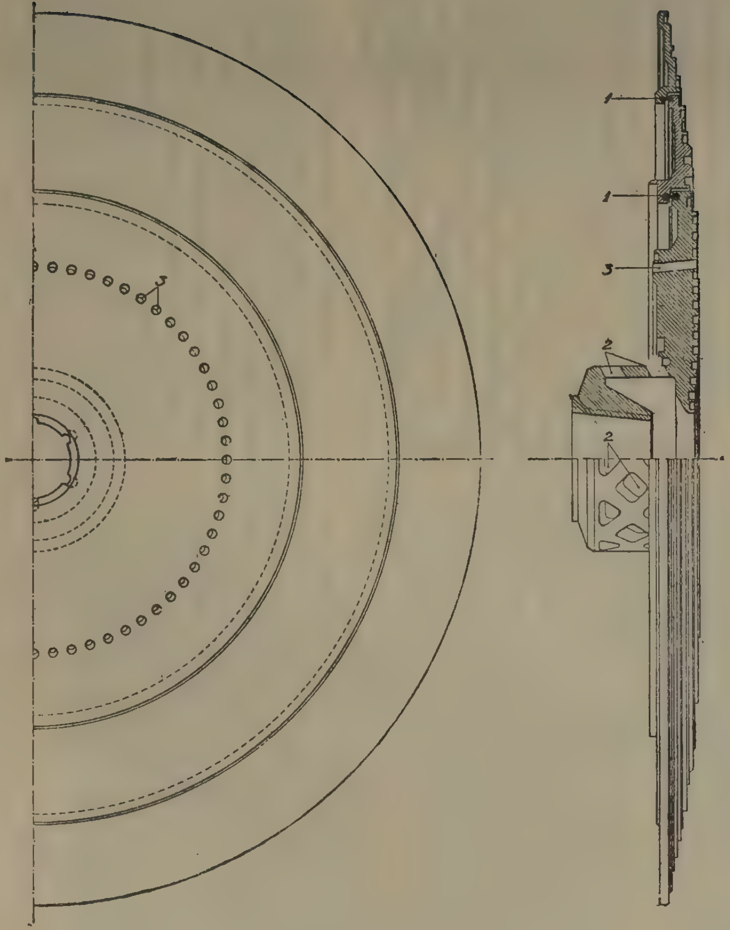

FIG. 51. LJUNGSTRÖM TURBINE DISC

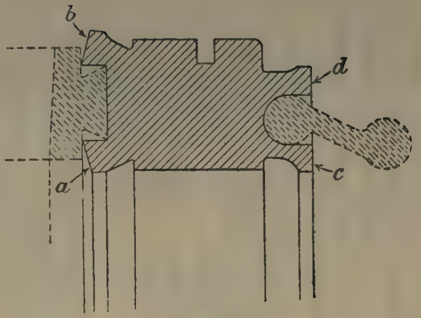

FIG. 50

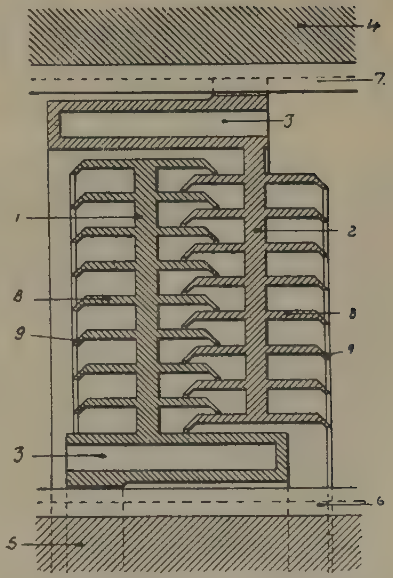

FIG. 52. LABYRINTH SHAFT PACKING FOR LJUNGSTRÖM TURBINE

to the rotor discs *F* and *G*. The attachment is made in each case by means of the above-mentioned expansion rings.

Leakage of steam from the steam chest is kept down to a very small amount by a most ingenious shaft gland *N*.

The steam pipe *A* is branched into two pipes *B* leading directly into the steam chest. The pipes *A* and *B* are surrounded by an air jacket enclosed in the space between the pipe and a light steel sheath, welded to the pipes. This jacket minimizes the loss by radiation to the exhaust steam by which the pipes are surrounded.

When the turbine is required to take an overload, an automatic by-pass valve or a hand-operated by-pass valve admits live steam to the blade system through the ports *L* and *M*. At these points, the blade area is greater than it is through the innermost ring of blades, so that the weight of steam flowing through the turbine per second is increased. There is, however, a slight reduction of efficiency at overloads.

The increased area for flow consequent upon the great increase in specific volume towards the exhaust end of the blade system is provided for partly by the increase in circumference and partly by increasing the axial length of the blades. There is, however, a limit to the safe length of the blades owing to the bending action of centrifugal force; and when the length required exceeds the safe length, the complete blade is divided into two or more sections, each section being attached to a continuous ring of steel, known as a "strengthening ring." An example of this is shown in Fig. 48. Here the last six rings of blades are subdivided into six sections.

DETAILS OF CONSTRUCTION

Blading. A section through a blade ring is shown in Fig. 49. The blades proper (11) are attached, in the first place, by welding, to two foundation rings (10). When the complete ring is formed, the foundation rings are turned to the dovetail section, as shown in Fig. 49. The "strengthening ring" (9), whose function is described above, is turned to the section shown in Fig. 50, the projecting edges *a* and *b* being rolled over so as to grip the dovetail of the foundation ring, and the edges *c* and *d* being rolled over to grip the expansion ring. The expansion ring (4) is gripped by a caulking ring (2), which is held firmly in a groove in the side of the rotor disc (1). Nickel steel fins (7), secured in the strengthening ring by caulking strips (8), run contiguous to the strengthening ring of the next outer group of blades and minimize the loss due to steam leakage.

The Rotor. In a radial-flow turbine of the Ljungström type, the central portion of the rotor discs is subjected to a high temperature, say, from 550° F. to 650° F., while the outer portions are in contact with comparatively cool steam at from 75° F. to 100° F., or thereabouts. Although heat is conducted from the centre of the disc outwards, through the metal, the central portion of the disc is usually much hotter than the outer portion, and a single disc would be subjected to high strains caused by the expansion and contraction. An example of the construction used in Ljungström turbines is given in Fig. 51. It will be observed that the disc is divided into three sections, which are attached to each other by the ubiquitous expansion ring (1). The steam passages in the hub of the rotor are seen at (2) and the overload steam admission ports at (3).

Shaft Glands. Reference to Figs. 46 and 48 will show that the shaft glands must withstand the full steam pressure, and, therefore, demand special care in design and manufacture. A section through a single element of the Ljungström gland is shown in Fig. 52, the complete gland being shown in Fig. 48. The rings for each element are turned out of solid steel, and are designed to expand freely in all directions without distortion. They are mounted on feathers (6) and (7).

The edges of the metal are turned very thin and rolled down to an angle of 45°, so that when the rings are in position, the bent edges make contact with the neighbouring rings. Actual initial contact is not harmful and, after the first few revolutions of the turbine rotor, a very slight clearance is worn. This clearance does not tend to increase materially.

The design is very satisfactory, insomuch that it is possible to arrange for about 160 throttlings of the steam from the initial pressure down to atmospheric pressure in an axial length of some 3 to 5 in.

APPLIED MECHANICS

By Geo. W. Bird, Wh. Ex., B.Sc., A.M.I.Mech.E., A.M.I.E.E.

LESSON XXV

Mechanics of Fluids—Flow of Water over Notches

In our explanation of *Bernoulli's theorem* and its application to the *Venturi meter* we have shown what a very simple and accurate instrument we possess in this meter for the purpose of measuring the rate of flow of fluids. Other methods of measuring fluid flow, however, may be employed, involving the use of notches or weirs. The shape of the notch is usually either triangular or rectangular. Mr. Francis, of Lowell, who made an extended experimental study of the flow through a sharp edged *rectangular notch*, deduced the expression

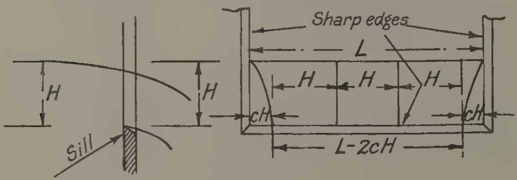

Fig. 159

$$Q = 3{\cdot}33\ H^{\frac{3}{2}}\left[L - \frac{n}{10}\ H\right]$$

Where Q = quantity flow through the notch in cub. ft. per sec.
H = still water head over the sill measured some distance back from the actual sill.
L = length of notch in ft.
n = number of end contractions = 0, or 1, or 2.

If L is the width of the notch, then it is seen that, with sharp edged sides as in Fig. 159, the flow may be considered as being made up of, (1) the flow through a number of square orifices of side H, and (2) the flow through the two (or 1, or 0, as may be arranged) end contractions. When L is large compared to H the width of the end contractions is unaffected by L and is found to be simply proportional to H, or, width of end contraction = cH. Now the flow through such an orifice is found to be proportional to $H^{\frac{5}{2}}$, and hence the total flow through the central rectangular portion is proportional to

$$\left(\frac{L - 2cH}{H}\right) H^{\frac{5}{2}}$$

or, $$Q_1 = K_1 \left(\frac{L - 2cH}{H}\right) H^{\frac{5}{2}} \quad . \quad . \quad . \quad (1)$$

The flow through the end contractions is proportional to $ncH^{\frac{5}{2}}$

or, $$Q_2 = K_2\ ncH^{\frac{5}{2}} \quad . \quad . \quad (2)$$

The total flow

$$Q = Q_1 + Q_2$$

$$Q = K_1 H^{\frac{5}{2}}\left[\frac{L - 2cH}{H} + \frac{K_2}{K_1} nc\right] \qquad K_3 = \frac{K_2}{K_1}$$

$$Q = K_1 H^{\frac{5}{2}}\left[\frac{L}{H} - 2c + K_3\, nc\right]$$

$$Q = K_1 H^{\frac{3}{2}}\left[L - (2c - K_3 nc)\, H\right] \quad . \quad (3)$$

The various constants involved can be simplified and determined experimentally and thus (3) may be expressed as

$$Q = 3{\cdot}33\, H^{\frac{3}{2}}\left[L - \frac{n}{10}\ H\right] \quad . \quad . \quad . \quad (4)$$

Example. Determine the flow through a sharp edged rectangular notch 4 ft. wide, the still water head over the sill being 18 in. There are two end contractions.

$$Q = 3{\cdot}33 \times (1{\cdot}5)^{\frac{3}{2}}\left[4 - \left(\frac{2}{10} \times 1{\cdot}5\right)\right]$$

or, $Q = 22{\cdot}63$ cub. ft. per sec.

Thomson's Principle of Similarity

This may be stated briefly as—

Assuming a frictionless fluid, then the flows through similar orifices similarly situated in similar vessels are proportional to

$$(\text{the linear dimensions})^{\frac{5}{2}}.$$

Let Fig. 160 represent any two similar vessels containing a frictionless fluid, and having similar orifices situated as shown; then, if the linear dimensions of the large orifice A are

n times the corresponding dimensions of the small orifice B and, also $H = nh$ (the orifices being similarly situated) then we may state—

Flow through A : flow through B :: $n^2 : 1^2$ and also, flow through A : flow through B :: $\sqrt{H} : \sqrt{h}$

$\therefore$ flow through $A \propto n^2 \times \sqrt{\frac{H}{h}}$ times the flow through B or, flow through $A \propto (n)^{\frac{5}{2}}$.

Thomson's *Principle of Similarity* enables us to state the form of the expression for the flow through a *triangular notch*. For any given triangular notch (Fig. 161) all cross-sections of flow for different heads are of similar triangular form, and hence we may write

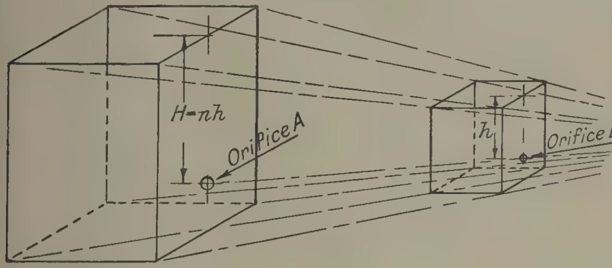

FIG. 160

$$Q \propto H^{\frac{5}{2}}\,; \text{ or, } Q = KH^{\frac{5}{2}}$$

where K is a constant to be determined experimentally and is usually taken as 2·635 when the inclination of the sides of the notch to the horizontal is 45°, that is, the angle of the notch is 90°. If the angle is other than 90°, the appropriate value of the constant must be used. Where the rate of flow is not too great the triangular notch is better than the rectangular since the coefficient remains more nearly constant with a varying head.

EXAMPLE. Determine the flow in gallons per minute through a 90° notch when the head is 9 in.

$$Q = 2{\cdot}635 \times (0{\cdot}75)^{\frac{5}{2}} \text{ cub. ft. per sec.}$$

or, $Q = 1{\cdot}283$ cub. ft. per sec.

$Q = \underline{479{\cdot}8 \text{ gals.}}$ per min.

FLOW THROUGH PIPES

The flow of a fluid through a pipe is always accompanied with losses, and one of these is that due to the friction between the particles of the water themselves and also between the water and the pipe surface. This loss, which is a *loss of energy*, is usually referred to as the *loss of head due to friction*. Other losses which also occur are due to sudden enlargements or contractions of section, sudden changes in the direction of flow and partial obstructions in the line of flow, and each of these may also be stated as a *loss of head*.

FRICTION LOSS. Prof. Osborne Reynolds has shown the existence of a *critical velocity of flow* and determined that, within reasonable limits of accuracy, this friction loss is directly proportional to the velocity of flow, provided that the velocity does not *exceed* a certain *critical value*. When, however, this critical value is exceeded turbulence is set up and the friction loss is then more nearly proportional to the square of the velocity.

An expression for the loss of head h which occurs, due to friction, when water passes through a pipe and the pipe is running full, is

$$h = \frac{4flv^2}{2gd}\,;$$

where f = a coefficient which is affected by the size and condition of the pipe.
l = length of the pipe.
d = diameter of the pipe
v = velocity of water through the pipe.

The value of f for cast-iron pipe of clean surface has been found to vary from 0·01 for a pipe of 1 in. diameter to 0·005 for a very large pipe, say, 2 ft. or more in diameter.

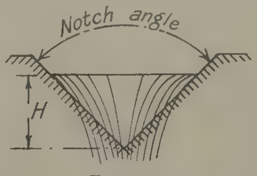

FIG. 161

EXAMPLE. Calculate the head lost in friction per mile for a clean C.I. pipe, 12 in. in diameter, when delivering 400 cub. ft. per minute. Assume $f = 0{\cdot}0054$.

To determine v we have

$$Q = A \times v$$

$$\therefore\quad v = \frac{400 \times 4}{60 \times \pi} = \underline{8{\cdot}487 \text{ ft.}} \text{ per sec.}$$

$$\text{loss of head} = h = \frac{4flv^2}{2gd}$$

or,
$$h = \frac{4 \times 0{\cdot}0054 \times 5280 \times (8{\cdot}487)^2}{2 \times 32{\cdot}2 \times 1}$$

$\therefore\ h = \underline{127{\cdot}6\text{ ft.}} = \underline{55{\cdot}5\text{ lb.}}$ per sq. in. (approx.).

LOSS OF HEAD DUE TO SUDDEN ENLARGEMENTS

When a pipe is suddenly enlarged a loss of head is incurred which is proportional to the square of the difference of the velocities at the two sections 1.1. and 2.2., Fig. 162. Let us assume that the pressure over the annular surface *BAAB* is the same as that at section 1.1.

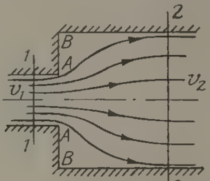

FIG. 162

Experiment shows that it is so. The momentum per second at section 1.1.

$$= \frac{a_1v_1w \times v_1}{g} = \frac{Qwv_1}{g} \quad . \quad . \quad (1)$$

Also, the momentum per second at section 2.2.

$$= \frac{a_2v_2w \times v_2}{g} = \frac{Qwv_2}{g} \quad . \quad . \quad (2)$$

Where a_1 = area of section at 1.1. v_1 = velocity of flow through section 1.1.

a_2 = area of section at 2.2. v_2 = velocity of flow through section 2.2.

W = weight per unit volume of water. $Q = a_1v_1 = a_2v_2$.

Now, we have previously shown that, to produce this change by momentum, we shall require a force action whose magnitude is given by

Force = change of momentum per sec.

or,

$$\text{Force} = \frac{Qwv_1}{g} - \frac{Qwv_2}{g} = \frac{Qw}{g}(v_1 - v_2)$$

We can now equate the forces which act upon the water between the two sections 1.1. and 2.2., Fig. 162, and we write

$$p_2a_2 = \frac{Qw}{g}(v_1 - v_2) + p_1(a_2 - a_1) + p_1a_1$$

or, $p_2a_2 = \frac{a_2v_2w}{g}(v_1 - v_2) + p_1a_2 - p_1a_1 + p_1a_1$

$$\therefore \frac{p_2 - p_1}{w} = \frac{v_2}{g}(c_1 - v_2)$$

or, $\frac{p_2}{w} = \frac{p_1}{w} + \frac{v_2}{g}(v_1 - v_2)$ = the pressure head at the section 2.2. . . . . . (1)

If we assume that no loss takes place between sections 1.1. and 2.2. we can apply Bernoulli's equation and write

$$\frac{p_1}{w} + \frac{v_1^2}{2g} = \frac{p_2}{w} + \frac{v_2^2}{2g} \qquad [h_1 = h_2]$$

$$\therefore \frac{p_2}{w} = \frac{p_1}{w} + \frac{v_1^2 - v_2^2}{2g} \quad . \quad . \quad (2)$$

Whatever loss has taken place due to the sudden enlargement will be given by (2) – (1)—or, loss of head

$$= \frac{p_1}{w} + \frac{v_1^2}{2g} - \frac{v_2^2}{2g} - \frac{p_1}{w} - \frac{v_2v_1}{g} + \frac{v_2^2}{g}$$

$$\therefore \text{ loss of head } = \frac{(v_1 - v_2)^2}{2g} \quad . \quad . \quad (3)$$

LOSS OF HEAD DUE TO SUDDEN CONTRACTION

This may be regarded as a case of sudden enlargement from section 1.1. to section 2.2., Fig. 163, the loss due to passing through the orifice in the plane *BB* being negligible, and we may now write

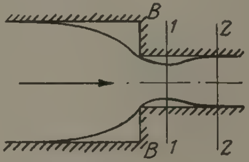

FIG. 163

Loss of head $= \frac{(v_1 - v_2)^2}{2g}$; and since the coefficient of contraction for a sharp edged circular orifice is about 0·6, it is usual to take this value, and the loss of head is then

$$\text{Loss of head} = \frac{\left(\frac{v_2}{\cdot 6} - v_2\right)^2}{2g} = \frac{(1{\cdot}666v_2 - v_2)^2}{2g}$$

or, loss of head $= \dfrac{0{\cdot}44v_2^2}{2g}$ (nearly), and as the exact value of the coefficient 0·6 is uncertain, it is usual to take the loss of head as $\dfrac{0{\cdot}5v_2^2}{2g}$.

HYDRAULIC GRADIENT

Suppose EX, Fig. 164, to represent a long pipe which is fitted with vertical tubes at a number of points along its length. Also, let there be a valve fitted to the pipe at X whereby the flow through the pipe can be controlled. At E, the entrance to the pipe, there will be a loss due to the sudden contraction, and this loss will be shown by the water in the first vertical tube, fitted to the supply tank, failing to rise to the level of the free surface in the tank by a height h_1. Again, it will be seen that the level of the water in the second tube is below that in the first tube by an amount h_2, and this distance, h_2, represents the loss of head due to friction experienced by the water over the length l_1. Similarly we shall find that at each successive tube the level of the water is below that in the preceding one by an amount which represents the loss of head, due to friction in travelling through the pipe between the two tubes. If, now, we draw a line joining the tops of the water columns in each of the vertical tubes, this line is termed the *hydraulic gradient* and, at any point, the angle between the hydraulic gradient and the horizontal is termed the *virtual slope* and is equal to $\dfrac{\text{loss of head}}{\text{length of pipe}} = \dfrac{h}{l} = \text{i}$.

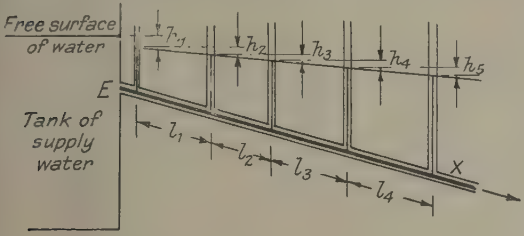

FIG. 164

For all ordinary cases of water flow through pipes i is a small quantity. This *hydraulic gradient* line is important and must be borne in mind if continuity of supply is to be secured. If the pipe at any point rises above the hydraulic gradient, the pressure at this point becomes less than atmospheric and air is liable to accumulate and interrupt the flow. In practice, it is desirable to keep the pipe line below the hydraulic gradient.

EXAMPLE. A 4 in. diameter pipe is suddenly enlarged to 8 in. diameter. The delivery through the pipe being 40 cub. ft. per min. determine the loss of head due to the sudden enlargement. If the direction of flow be reversed, calculate the loss of head.

$$Q = A \times v_1$$

$$\therefore\ v_1 = \frac{Q}{A} = \frac{40 \times 3 \times 3 \times 4}{60 \times \pi}$$

$$= \frac{24}{\pi} \text{ ft. per sec. through 4 in. pipe,}$$

and, $v_2 = \dfrac{6}{\pi}$ ft. per sec. through 8 in. pipe.

$$\therefore\ \text{loss of head} = \frac{(v_1 - v_2)^2}{2g} = \frac{\left(\frac{24}{\pi} - \frac{6}{\pi}\right)^2}{64{\cdot}4}$$

$$= \underline{0{\cdot}5097 \text{ ft.}}$$

Flow Reversed

$$\text{Loss of head} = \frac{0{\cdot}5 \times \left(\frac{24}{\pi}\right)^2}{2g} = \underline{0{\cdot}4530 \text{ ft.}}$$

EXAMPLE. A 15 in. water main having a virtual slope of 0·004 conveys water at 4 ft. per sec. Determine the value of the coefficient f and the friction resistance per foot length of pipe.

$$i = \frac{h}{l} = \frac{4fv^2}{2gd}$$

$$\therefore\ f = \frac{2gdh}{l4v^2} = \frac{2 \times 32{\cdot}2 \times 15 \times 4}{4 \times 16 \times 12 \times 1000}$$

or, $\quad f = 0{\cdot}00503.$

Loss of head per foot length of pipe

$$= i = \frac{4}{1000} \text{ ft.}$$

$\therefore$ loss of head per foot length

$$= \frac{4}{2{\cdot}3 \times 1000} \text{ lb. per sq. in.}$$

and this pressure acts over the area of cross-section to overcome the frictional resistance,
$\therefore$ frictional resistance per foot length

$$= \frac{4}{2{\cdot}3 \times 1000} \times \frac{d^2\pi}{4}$$
$$= 0{\cdot}3073 \text{ lb.}$$

(CONCLUSION)

POWER TRANSMISSION BY CHAINS

By H. M. Andrew, Wh.Ex.

LESSON III

THE THEORY OF CHAIN GEARING

The following abbreviations are made use of in this section. Assume—

A = bearing area of a single chain bearing, in sq. in.
B = bearing pressure, in lb. per sq. in.
C = centrifugal tension in lb.
G = gravity constant = 32·2.
H = horse-power transmitted.
N = number of teeth in a chain wheel.
P = pitch of chain, in inches.
R = revolutions per minute of a wheel.
S = linear speed of a chain, in ft. per min.
T = tension in a chain, due to H, in lb.
W = weight of a chain, in lb. per ft. of length.
Z = the limiting speed, in ft. per min.

The linear speed of a chain $S = \frac{RNP}{12}$ ft. per min.

Centrifugal Tension. When any transmission such as belts, ropes or chain is run at speed around wheels a centrifugal tension (C) is set up. This acts as a pressure on the chain bearings which equals $\frac{WS^2}{3600g}$ lb.

The horse-power to be transmitted from wheel to wheel sets up an additional load (T) on the driving side of the chain equal to $\frac{H \times 33{,}000}{S}$ lb. The working load on the driving side of a chain transmitting power is therefore equal to

$$T + C = \frac{H \times 33{,}000}{S} + \frac{WS^2}{3600g} \text{ lb.}$$

and the chain must be strong enough to withstand this load, with a sufficient factor of safety.

Bearing Pressure. The pressure per square inch on the chain bearings due to the above load $= \frac{T + C}{A}$.

In discussing bearing pressures, etc., A will be assumed to have its fully developed catalogue value, although, as explained in Lesson II, this is often not developed until the chain has been in service for a while. The bushes and pins of bush roller chains may have slight distortions or irregularities that reduce the actual bearing until the surfaces have worn together. In the case of inverted tooth chain, inequalities in pitch of the links forming any particular pitch of the chain have also to be equalized by wear before all the links are fully bearing.

In both these cases accuracy of manufacture is a decided advantage but, as the cost of making an initially perfect chain would be prohibitive, some slight final bedding down always occurs in service. As the early bearing pressures are undeterminable, the assumed bearing pressure is always taken as that which would occur on a fully bedded-in bearing.

The full bearing pressure comes on all the bearing pins on the driving side of the chain, but, in a smooth drive, only those pins entering or leaving chain wheels are subject to any wear by rotation in their bearings. Flapping or vibration of the chain, of course, causes additional wearing points. When a bearing pin enters the wheel teeth and moves round with the wheel, the pressure on this pin is gradually reduced until it comes to a point on the wheel where no bearing pressure exists. The position of this minimum pressure point varies with the horse-power and speed of the chain.

First quality bush roller chains, with good lubrication, can carry a bearing pressure as high as 4,000 to 4,500 lb. per sq. in. The design of inverted tooth chain does not admit of more than, say, 3,000 to 3,500 lb. per sq. in. being practicably employed. Driving chain makers regard 2,000 lb. per sq. in. as about the desirable maximum that will secure a reasonable chain life for the average industrial drive.

Pressure on Wheel Teeth. The chain load is carried by several teeth in the wheel. The thrust on any particular tooth is the difference between the load on the bearing pins that are on either side of it. A wheel with a large number of teeth distributes the load over a greater number of teeth, of course, than a wheel with a smaller number of teeth.

Apart from friction, the thrust is normal to the surface of the tooth face.

Polygonal Effect. The impact between chain and wheel caused by polygonal action was described in Lesson I. The loss of energy due to this impact is negligible, and has no appreciable effect on the mechanical efficiency of the drive. The amount of noise created, however, varies in an inverse ratio to the number of teeth in the chain wheel and, consequently, wheels with a large number of teeth are much quieter than wheels with few teeth.

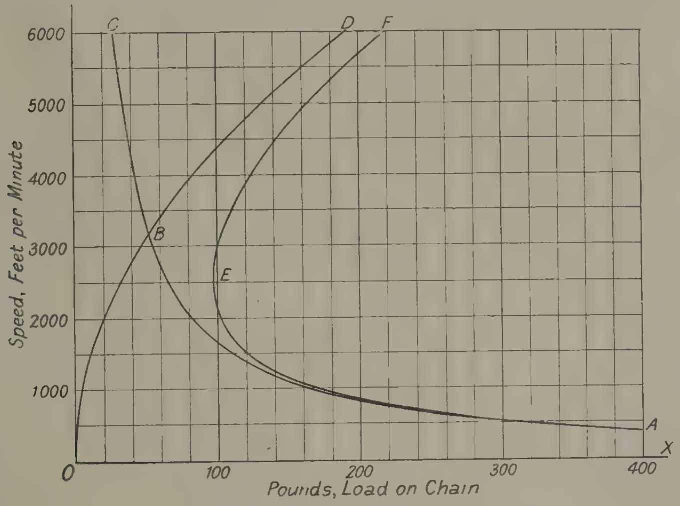

FIG. 13. CHAIN PERFORMANCE CURVES

Flapping of the chain due to any cyclic irregularity, of course, increases the impact and noise enormously, and should therefore be prevented as far as possible by close adjustment or the use of jockey wheels.

Catenary Pull. The natural catenary sag of a chain between the chain wheels is somewhat interfered with by the driving tension T on the tight side. The catenary pull is, therefore, best calculated from the slack side and regarded as an additional load on the chain bearings. Its effect is negligible except in the case of long horizontal drives using large pitch (and therefore heavy) chain, or in cases where both sides of the chain are drawn taut.

A nearly horizontal chain assumes a parabolic shape, and we may assume it to be carried at wheel points P and P_1, whose *horizontal* distance apart is Y. Then, with a sag G at the lowest point O, the pull in pounds at point O $= \frac{WY^2}{8G}$, where Y and G are both in feet.

And the pull at points P would $= \frac{WY^2}{8G}\sqrt{1 + \frac{16G^2}{Y^2}}$

In the majority of cases, however, the slack side may be regarded as a catenary curve, when the catenary pull at the lowest point $= \frac{WY^2}{8G}$, and the maximum pull, at $P = W\left(G + \frac{Y^2}{8G}\right)$

In the case of an inclined drive, divide the span into two parts at the lowest point of the sag, and calculate for each section separately, doubling the horizontal length of each section to obtain Y.

Chain Performance Curves. For any type of driving chain of given pitch and weight per foot, a series of curves can be drawn to represent graphically the load and linear speed at which various horse-powers can be transmitted by this particular chain.

A sample curve of this type is illustrated in Fig. 13 where the horizontal abscissae OX represent the total load pull $T + C$ in pounds on the chain, while the vertical ordinates OY

equal the linear chain speed S in feet per minute. The curve ABC is representing the value of T for a particular horse-power at the various speeds, and the parabolic curve OBD represents the increasing centrifugal tension C due to the weight of this particular chain, as the linear speed increases.

Curve AEF is the performance curve for this chain in delivering the particular horse-power in question, and is formed by adding the horizontal values of the curves ABC and OBD together.

The horse-power under consideration can be obtained from this chain by running it under the conditions obtaining at any of the points in the curve AEF, either at low speed with a heavy load on the chain bearings, or at medium speed with a light load or at high speed with a heavy load due to centrifugal tension.

It is obvious that the minimum load on the chain bearings in delivering this particular power would occur at the speed corresponding to point E, and this speed is referred to as the limiting speed Z. Speed Z in feet per minute $= 1240 \sqrt[3]{\frac{H}{W}}$, and at this speed the centrifugal tension C is exactly half the value of the load tension T. In making use of such a curve it must be kept in mind that the allowable bearing pressure in pounds per square inch permitted by the chain maker must not be exceeded and the effect of increased speed on the noise created must also not be overlooked.

It is never advisable to work the chain at any point on the curve AEF above the limiting speed point E, as this causes wasteful wear by centrifugal tension, without any compensating work done to show for it. The best point to deliver this particular horse-power with such a chain is at about one-half to three-quarters of the limiting speed, where noise is minimized and mechanical efficiency is at a maximum.

The Special Case of Rocker Joint Chains. The rocker joint type of chain has, of course, no "bearing area" in the usual sense, and for a given load the pressure between the rocker and its seat pin is governed by different considerations to those which limit the pressures employed in the case of cylindrical pin bearings.

A theoretically perfect knife edge bearing would have no friction, and crushing would tend to occur rather than wear. The manufacturing difficulties of this design obviously consist in the endeavour to produce pure rocking with a minimum of sliding of the rocker across its seat pin, and a great many improvements have been incorporated from time to time to secure this end. Apart from the nature of its bearing load, this type of chain is, however, governed by the same considerations that apply to the pin bearing type of chains, and its performance curve can be limited in a similar way by the maximum permissible pressure allowed in the bearing.

Fatigue in Chain Drive. The valuable data that has been obtained from time to time by scientific investigators on the subject of fatigue of materials is of particular importance in the case of power transmissions, which are naturally subject to a variety of intermittent stresses from one cause or another. A power transmission offers such a complex problem from this aspect that it is impossible in the present state of knowledge on this subject to do other than simply cover against this aspect in the factor of safety employed in the drive.

At present it seems inadvisable to use a factor of safety of less than 10 in the case of bush roller chain and, say, 15 in the case of inverted tooth chain, and where shock load is obviously present these factors should, when possible, be increased by 50 to 100 per cent.

STRUCTURAL STEELWORK

By W. Arnold Green, M.A., B.Sc., A.M.Inst.C.E., M.I.Struct.E.

LESSON VII

Columns and Struts. It sometimes happens, as in the case of stanchions carrying light roofs subject to heavy wind loads, that the bending stresses are greater than the axial stresses, that is the stanchion is more a beam than a strut.

In such cases it is unreasonable to use the low stresses permitted for the member considered as a strut, and the method suggested in the 1921 Belgian standard specification seems reasonable.

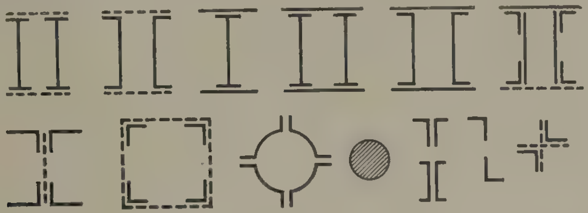

Fig. 34

By this specification, columns are calculated for the allowable maximum stress permitted in beams, for *reduced areas*, i.e. the actual column areas reduced in the proportion of the buckling stress to the ultimate breaking stress on which the allowable maximum stress is based. For combined direct compression and bending, the stresses are calculated independently, and when added must not exceed the allowable maximum. This is equivalent to the following treatment—

If fa = actual bending stress allowable in a column.

pa = actual axial stress allowable in the column at the same time.

f = allowable bending stress in a beam.

p = allowable axial stress in the column with no bending.

$$\text{then} \quad \left(pa \times \frac{f}{p}\right) + fa = f$$

$$\text{i.e. } pa = (f - fa)\frac{p}{f} \quad \text{or } fa = f\left(1 - \frac{pa}{p}\right)$$

If applied to the example at the end of the last article, $f = 7\frac{1}{2}$, $p = 4{\cdot}02$, $fa = 1{\cdot}46$, $pa = 6{\cdot}04 \times 4{\cdot}02 \div 7\frac{1}{2} = 3{\cdot}24$ instead of 2·59, i.e. the central load could be 62½ tons instead of 50 tons, or with $pa = 2{\cdot}59$, $fa = 7\frac{1}{2}\,(1 - {\cdot}64) = 2{\cdot}71$ instead of 1·46, i.e. a bending moment of 167 in. tons instead of 90 in. tons could be sustained.

Unfortunately, in addition to the uncertainty as to suitable column stresses, there is no such direct method for calculating columns as there is for beams, and suitable sections can be selected only by the method of trial and error guided by experience.

In Fig. 34 are shown in section a few of the many possible types of compression members other than plain joists or broad flange beams.

The dotted lines signify either lattice or isolated plate connections between the main members.

Solid round columns are not economical, but are used where minimum dimensions are essential. Angles and tees are largely used in roof trusses. Pairs of channels are often employed for large truss rafters which have to act as beams as well.

In a built-up compression member, it is important to ensure that the component parts are at least as strong as the member as a whole. For instance, plates riveted to the flanges of a built-up stanchion must be held by rivets at sufficiently close a pitch to enable them to act as independent struts with ends fixed, capable of carrying and with the same factor of safety the maximum stress coming on the stanchion flange as a whole.

From the strut formulae given in B.E.S.A., No. 153, the maximum length of a strut for the maximum allowable stress is $45\frac{1}{2}k$ if the ends are fixed and $30k$ if pin connected.

The radius of gyration (k) of a plate of thickness t is $t/\sqrt{12}$, so that a plate riveted to the flange should be as strong as the flange if the length between rivets is $45\frac{1}{2} \times t/\sqrt{12} = 13t$.

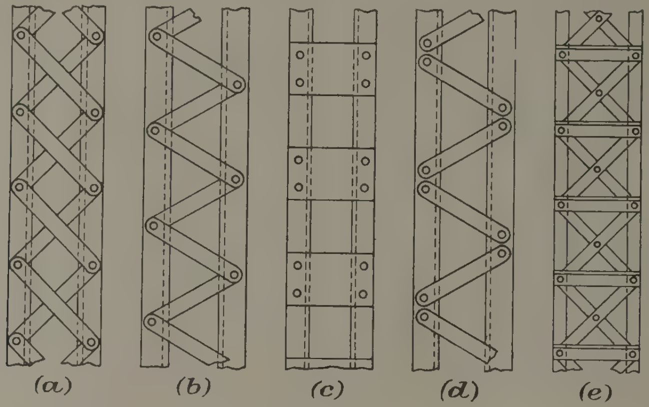

Fig. 35

In a compression member held at one or more points along its length it may be possible for bending to take place in a wavy line, so that the lengths between two adjacent points of restraint must be regarded as pin connected. This may be the case in a roof rafter, or top boom of a girder, where, in the plane of the truss, the rafter or boom may be considered as fixed at the ends, but for resisting buckling in a direction at right angles to this plane pin connections must be assumed.

Compression members of large gyration radius can be formed by fastening together two joists or channels or four angles by means of lattice bars made of flats or angles.

Tests made by Professor H. F. Moore, of Illinois University, to determine the efficiency of such bracing in making the two connected members act as one, demonstrated that cross bracing at 45° (Fig. 35(*a*)) is more efficient than single bracing at 60° with the axis (Fig. 35(*b*)), which, in turn, is more efficient than batten plates (Fig. 35(*c*)), and single bracing in which there is not proper triangulation (Fig. 35(*d*)). In the experiments, stresses were developed in the lattice bars which would have been produced by a transverse shear 1 to 3 per cent of the applied axial load, but it seems impossible to predict the probable stress in them. It is usual to make them of flats of a width about three times the rivet diameter, and of a thickness about one-fortieth of their length for single lacing, or one-sixtieth for double lacing where they are riveted at their intersection.

It is not advisable to adopt a system such as that illustrated in Fig. 35(*e*), which is itself capable of transmitting stress independently of the connected members as the rivets

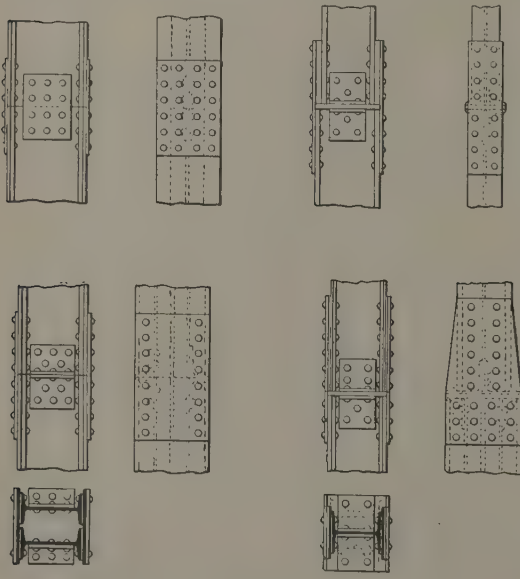

Fig. 36

may be stressed to a much greater extent than intended.

Joints in compression members are usually made with splice plates. If the butting ends are machined, and care is taken to ensure that they are in contact, it is often not necessary, as is the case with tension members, for the splice plates to have the same area as the member, and for the rivets on each side of the joint to be capable of transmitting the whole of the load.

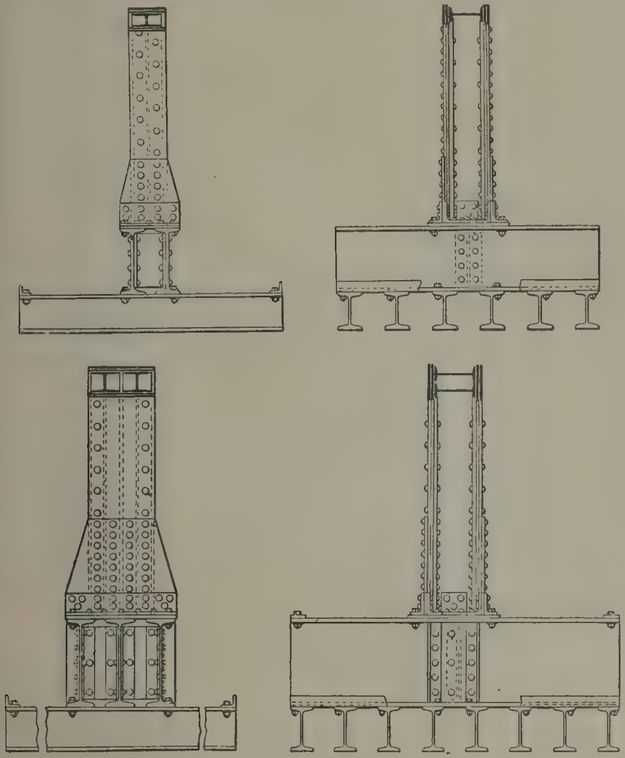

Fig. 37

In building construction it is now common practice to splice the columns just above the floors as indicated in Fig. 36.[1] It is advisable to ensure that such splices are able to resist as much bending moment due to eccentricity and (or) wind as is assumed to be resisted by the column below the floor.

The allowable pressure on the subsoil depends on the nature of the latter. Fifteen tons or more per sq. ft. may be permitted on a rock foundation. It is common practice to allow 4 tons per sq. ft. on compact gravel, London clay, or chalk and less pressures for ordinary clay. To provide a sufficient area under a stanchion, the latter is usually placed on a thick block of concrete. When a large area is required it may be necessary to provide a reinforced concrete footing, or to place the stanchion on two or more tiers of joists surrounded by concrete forming a *grillage* foundation. To spread the column load on the concrete a stiff base plate of sufficient area is fixed to the bottom of the column by means of gusset plates and angles, such as are illustrated in Fig. 37.[1] The L.C.C. regulations require that the total load must be taken through the rivets connecting the column shaft to the base, but in the case of a grillage foundation, where the shaft stands on the top tier of joists, only a portion of the load depending on circumstances need be taken through the rivets. Each tier of grillage joists is designed as a double cantilever, the total bending moment resisted in each tier being usually taken as the product of the total load carried, and one-fourth of the length of the projecting portion.

The last lesson will be a brief discussion on roof trusses.

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[1] Figs. 36 and 37 are taken from *The Handbook for Constructional Engineers*, issued by Messrs. Dorman Long & Co., Ltd.

BLAST FURNACE PRACTICE

By Percy C. R. Kingscott, D.I.C., A.R.C.Sc., F.I.C., B.Sc. (Lond.)

LESSON XII

FURNACE BURDENING

The burdening of a furnace is the determination of the proportions of ore, limestone, and coke in order to produce the required kind or grade of pig-iron, at the same time ensuring the successful working of the process. It is necessary to know the physical and chemical characteristics of the materials available and, also, to consider the manner in which the principal elements associated with the iron can be regulated, for their percentage content determines the class of pig, viz., foundry, basic, low phosphoric, forge, high silicon, Bessemer, and other varieties. These elements are silicon, sulphur, phosphorus, manganese, and carbon.

Control of Silicon. The percentage of silicon entering the iron depends upon the temperature of the hearth and the analysis of the slag. The hearth temperature is raised by increasing the blast temperature, or by lowering the ratio of ores to fuel. A limey slag has a strong affinity for silica, and will remove it before it can be reduced to silicon. Conversely, with the same temperature, a slag containing less basic material will allow silicon to enter the pig.

Lime tends to increase the temperature of the slag, as such additions render the slag more infusible, and thereby increases the temperature of the pig-iron. Thus, there are actual conditions when increasing the lime above a certain percentage may increase silicon in the pig, the chemical effects being masked by the influence of the temperature. Alumina, by rendering the slag more infusible, also increases the temperature, and this compound is desirable on that account where silicon is required, for example, in foundry irons.

Low silicon iron for basic steel manufacture is made by increasing the weight of the burden in the presence of a basic slag; by a heavy burden combined with high blast temperature and a less limey slag, or by a lighter burden, a limey slag, and moderate blast temperature, all these methods necessarily being compatible with efficient desulphurization. It is found that magnesia replacing lime has a beneficial effect upon basic iron slags, in that respect being the converse of the effect of alumina. Generally, an increase in silicon accompanies a decrease in sulphur and vice versa.

Control of Sulphur. Charcoal furnaces are generally free from trouble, due to this element, which indirectly produces many of the difficulties associated with coke practice. As has been stated, coke contains varying quantities of sulphur and, as pig-iron consumers usually set a maximum content of about ·06 per cent, it is evident that the slag must remove the majority of this element. There are several methods of achieving this result: increase in temperature, increase in volume of slag, increase in lime content, increase in manganese, and increase in the fluidity of the slag.

Economical practice is ensured by using a fluid and not too basic slag containing some manganese. Irons which must contain silicon can use less fluid slags, and obtain desulphurization by means of high temperature. Increasing the basicity, temperature, and volume of the slag naturally sacrifice full economy.

Generally, an increase in sulphur denotes a cold hearth due to water trouble, or a slag deficient in basic material.

Control of Phosphorus. Unless the furnace be deranged, all the phosphorus enters the pig-iron. Thus, when making low phosphorus pig it is extremely difficult sometimes to select the ingredients of the burden, which, working together, will produce the required specification. Naturally, such iron commands a premium in price.

Control of Manganese. As with silicon and sulphur, the manganese entering pig varies with the slag composition and its temperature, and it is impossible to secure its entire elimination in a normal furnace, or to reduce more than a certain percentage of the total present. Manganese is more active chemically than iron, and tends to form manganese silicate in the slag. It has a higher heat of reduction than iron and requires a higher temperature for that action. Hence, in order to reduce the largest amount, the slag must be limey in order to prevent the formation of manganese silicate, and of a high temperature, the latter being generally achieved with a light burden. On the other hand, a siliceous slag and cool hearth tend to lower the manganese content of the iron. Such slags have a cutting effect upon the brickwork of the hearth, owing to the manganese in the slag attacking the silica of the brick. At the same time, partially reduced material may descend into the hearth, and the resultant slag containing a large quantity of ferrous silicate, has a marked scouring effect upon the hearth walls.

Control of Carbon. The analysis of the burden has no effect upon the quantity of carbon entering pig-iron, although conditions of high temperature, associated with a large amount of carbon in the hearth, will produce a greater percentage of carbon in the pig. The total carbon in pig is fairly constant, and ranges from about 3 to 4 per cent, whether the iron be grey or white in fracture. This carbon exists in two forms, as free graphite and as combined carbon, the colour of the pig being due to the presence or absence of the former. Combined carbon is low in foundry iron and high in white basic irons.

Irons low in total carbon are made in furnaces working with low blast temperatures, large slag volumes, and fusible slags.

Silicon and phosphorus have an excluding effect upon carbon; on the other hand, manganese has an attraction for it.

Silicon tends to produce graphitic carbon at the expense of the combined carbon, as exemplified in grey foundry irons, which have a dark and coarse grain. It is interesting to note, however, that a hot calcareous slag will produce an iron with a grain far more open in appearance than would be expected from its silicon content.

Pig Fractures. Pig used for steel making is chosen on account of its analysis, the physical appearance being a secondary matter. With foundry irons, however, the appearance of the fractured surface is considered important, particularly in England. The appearance bears a relationship to the analysis although, as was mentioned in the previous paragraph, this is not always correct. Generally, a grey fracture denotes the presence of silicon and a close white grain that of sulphur.

The rate of cooling has a profound influence upon the appearance of the fracture, slow cooling favouring large grain. For this reason, foundry irons are made on sand pig beds, and allowed to cool without the use of water sprays. On the other hand, foundry iron is made successfully in America in chills or casting machines.

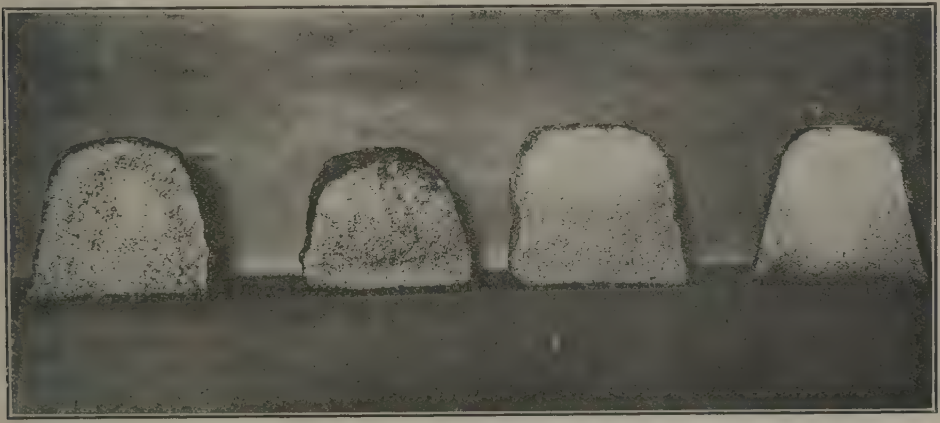

FIG. 35

Pig high in silicon has a glazed appearance when fractured and, at the same time, the tops of the pigs are generally rounded. High sulphur is denoted by a surface with a worm-eaten appearance and an otherwise "dirty" skin. The presence of manganese gives a crystalline appearance to the fracture and known sometimes as a spiegel fracture.

Fig. 35 shows, left to right, (1) foundry, (2) foundry, (3) foundry, and (4) forge fractures.

Fig. 36 shows, left to right, fine grey, mottled, white (basic quality), and spiegel white.

SLAGS

Our knowledge of slag is largely empirical, but we know that, far from being an undesirable nuisance in the extraction of iron, it accommodates all those non-volatile substances not required in the pig. Slags are calculated to produce certain results which have been suggested by experience, and not according to chemical formulae.

Blast furnace slags consist essentially of calcium silicate where either radicle is partially replaced by other radicles. Substances, such as lime, magnesia, alumina, and the alkalis enter the slag exclusively, as they are not reduced in the furnace. The lime of the slag is usually partly replaced by magnesia, small quantities of manganese oxide, alkalis, and iron oxide. The acid radicle is associated with alumina and sulphur, the latter in the form of calcium sulphide. Thus, at the outset, the quantity of lime, magnesia, alumina, and the major portion of the silica can be calculated from the mixture of ingredients, as these constituents, with the exception of alumina, are of definite chemical character. Alumina has been the cause of much argument as to whether it acts in an acid, neutral or basic capacity. Johnson states that, in all probability, alumina acts as a perfectly neutral substance being a diluent affecting the viscosity to some extent, but with a given ratio of lime to silica, not affecting the chemical nature of the slag. This theory has many adherents. It is clear that, without a knowledge of local practice, it is possible to treat alumina as an acid, and, giving the usual ratio of lime to acid, to produce so limey a slag as to "gob" the furnace. In this connection, it is safe to calculate the basicity of the slag as the ratio of bases to silica, as this ratio appears to be fairly constant in varying practice. Alumina increases the free running temperature of the slag without increasing its basicity and, for this reason, as stated before, is a suitable component of a slag when making foundry irons.

High alumina and magnesia together produce an extremely viscous and refractory slag.

A suitable furnace slag is perfectly fluid at the normal temperature of the furnace, so that it can be drained easily from the furnace. Such slags, hot, fluid, and grey, flow rapidly without bubbling or frothing.

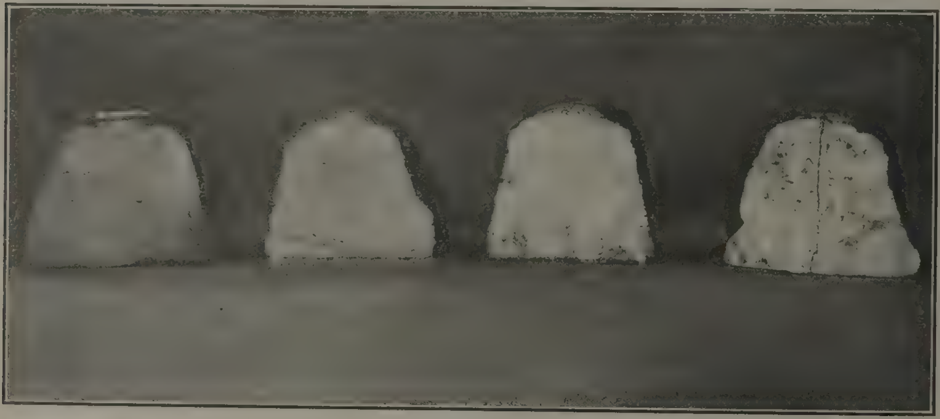

FIG. 36

The presence of alumina in basic iron slags is considered detrimental if in the range of about 18 to 22 per cent. Slags with the latter content of alumina do not permit of desulphurization and the simultaneons production of low silicon iron. H. E. Wright considers that alumina in slag acts as the outward sign of increased gangue, and is blamed for the effects of this increase, notably in Northampton and Cleveland practice. Keenan has stated that furnaces run smoothly with slags containing over 20 per cent of alumina. McCaffery has shown that there are 17 compounds of silica, alumina, lime and magnesia that enter into the composition of slags, in addition to the simple constituents themselves. He states that, whereas furnace authorities usually add the chemical equivalent in lime of the magnesia present to the lime, this is wrong practice as the magnesia minerals present in the slag are quite different from the non-magnesia substances. He also shows why a comparatively small change in the ultimate composition of the slag gives operating results which vary greatly. The physical properties of the components determine the physical properties of the slag, and a slight variation in the percentage of one or more constituent minerals will change the operating results of the furnace. In other words, it is the variation in the proportions of the compounds of the four primary constituents and not the amount of any individual one present which changes the slag properties.

Most slags fall within the following limits—

SiO_2	Al_2O_3	CaO	MgO	FeO	S
25–50	5–25	20–50	Tr.–25	Tr.–2	Tr.–2 per cent

It will be noticed that iron is lost in the slag as oxide, but other losses may occur as fine shots of metallic iron suspended in the slag. This latter can be recovered by treating the crushed slag in a magnetic separator, but the loss as oxide is final, this occurring chiefly in dark coloured lean slags. Slags from furnaces in trouble can contain as much as 8 per cent of iron as ferrous silicate.

THE TESTING OF PRIME MOVERS

By R. Royds, M.Sc., M.I.Mech.E.

LESSON VIII

STEAM-ENGINE AND STEAM TURBINE TESTS

The observations to be recorded in a steam-engine or turbine test depend mainly upon the purpose of the test. Very rarely is a reciprocating steam-engine tested merely to determine the mechanical efficiency, but if this efficiency were to be obtained, indicator diagrams would be taken from each end of each cylinder at definite intervals of time. Unless the engine were driving an electrical generator some form of brake would be necessary to load the engine and to determine the brake horse-power, or in marine engines on board ship a suitable torsion meter might be used to measure the shaft horse-power transmitted. The mechanical efficiency of a steam turbine cannot be measured directly as there are no means available for measuring the actual work done by the steam on the turbine blades. Thus, the shaft horse-power is the only direct measure of power which can be obtained from the turbine. If the turbine drives an electrical generator the shaft horse-power can be deduced from the electrical power generated divided by the efficiency of the generator at the tested load. Otherwise, a brake would be required or, on board ship, a torsion meter.

Measurement of Steam Consumption. Unless it is a case only of showing that the engine or turbine is capable of giving the guaranteed power, the tests are usually made to determine the steam consumption and thermal efficiency under stated conditions. The steam consumption is best measured by condensing the exhaust steam in a surface condenser, and measuring or weighing the discharge from the condensate pump. Allowances might be necessary for water or steam drained or bled from intermediate stages or jackets, or for leakage of condensate at the condensate pump. If steam from auxiliaries and the like entered the main condenser this should be allowed for, as well as leakages of sealing water, if used, into the condensate pump. It is also possible for dry air pumps to carry away an appreciable amount of vapour and small drops of condensate along with the air.

The tubes of surface condensers sometimes allow circulating water to leak to the steam side, and it is generally advisable to test for such leakage. The usual method is to keep the circulating pumps, air, and condensate pumps running after the engine or turbine has closed down. If the condensate discharge practically ceases within, say, twenty minutes or so after closing down, this is usually taken as an indication of tight condenser tubes. For this test it is usually best to close the main exhaust sluice valve, if such a valve is available.

The most accurate method of measuring the condensate is to discharge it into a tank on a weighing machine. For tests of long duration two similar tanks on weighing machines may be necessary and used alternately. Another arrangement is to use measuring tanks similar to those referred to in Fig. 38 (Lesson VII). In power stations, it is a common practice to have continuous records of the condensate discharge by using, say, a Lea Recorder or a Kent Venturi meter. The steam consumption per indicated horse-power hour (reciprocating engines) or per brake horse-power hour is obtained by dividing the total steam consumption per hour by the respective power values. If the engine drives an electrical generator the steam consumption per kilowatt hour is similarly determined. When tests are made for purposes of guarantee the conditions of the steam pressure, superheat, vacuum, load, etc., are usually specified. Under actual test these conditions may not be exactly fulfilled, and agreed upon allowances have to be made to correct the results to the standard or specified conditions.

Thermal Efficiency. The thermal efficiency of a steam-engine or turbine refers to the ratio between the amount of heat equivalent to the work done and the heat contents in the steam at the stop valve, reckoned from water at the temperature—pressure conditions existing at the exhaust. This may be based on the work done at the engine pistons in reciprocating engines, or on the work delivered at the shaft, or on the electrical energy delivered at the generator. The heat equivalent of the work done at the engine pistons is i.h.p. × 2,546 B.Th.U.'s per hour, and at the shaft b.h.p. × 2 546 B.Th.U.'s per hour. The heat equivalent of the delivered electrical energy is kw. × 3,412 B.Th.U.'s per hour. The heat contents of the steam in B.Th.U.'s per hour is the weight of steam used in pounds per hour multiplied by the heat contents per pound at the engine stop valve as given by Callendar's steam tables, less the amount of heat in water at the saturation temperature of the exhaust steam, both reckoned in B.Th.U.'s If steam is bled from the engine or turbine at intermediate stages, due allowance has to be made for the heat contents of this steam leaving the engine or turbine. Similarly, if steam or heat is supplied at intermediate stages, or to steam jackets, an allowance is necessary.

The ratio of the actual thermal efficiency to the theoretical efficiency of the Rankine cycle operating between the same conditions of steam pressures and temperatures is commonly termed the "efficiency ratio." Some prefer to term this ratio the "thermodynamic efficiency."

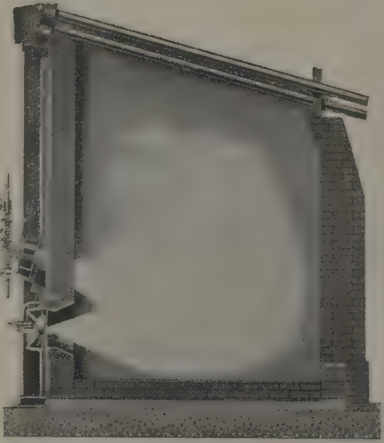

FIG. 70. SECTION OF FURNACE FITTED WITH OIL FUEL BURNERS

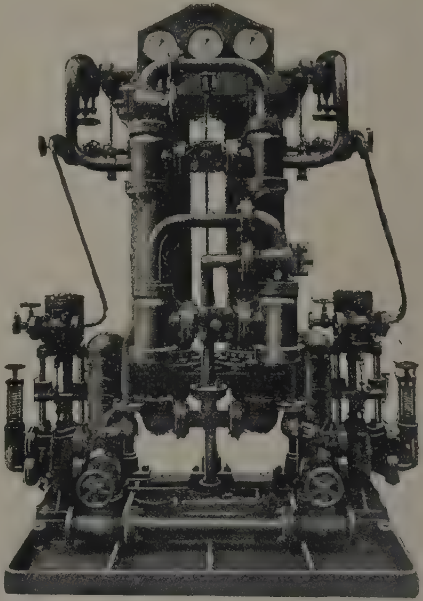

FIG. 72. COMBINED DUPLEX OIL PUMPS, HEATERS, AND FILTERS (PRESSURE SYSTEM)

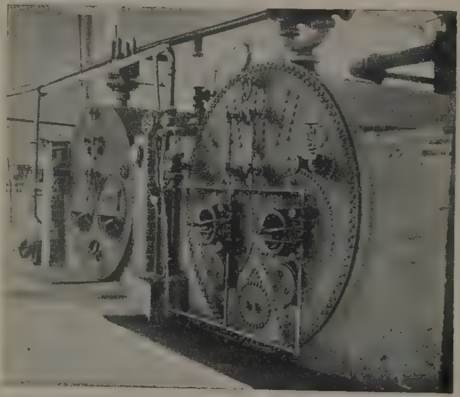

FIG. 71. LANCASHIRE BOILERS FITTED WITH OIL BURNER, USING STEAM AS ATOMIZING AGENT

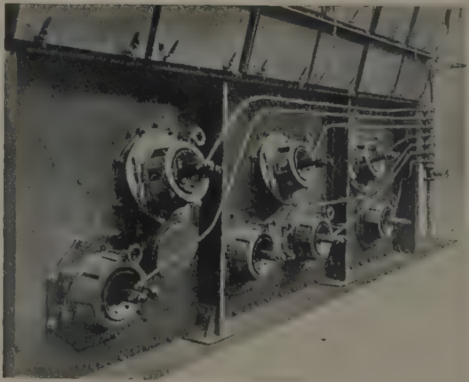

FIG. 73. FRONT OF BOILER FURNACE FIRED WITH OIL FUEL

(See Article on "Steam Boilers," page 1454)

FUEL CALORIMETERS

By Professor G. E. Scholes, M.B.E., M.Sc., M.I.A.E., M.I.Mech.E.

PART I

INTRODUCTION

Many substances when exposed to oxygen combine with it and produce oxides. This combination, as a rule, is accompanied by the liberation of heat, and if the conditions are favourable it takes place rapidly. Under such circumstances the substance is said to burn, and the heat liberated is such that the temperature attained by the burning substance is very high. Such heat may be used for commercial or other processes.

Rapid oxidation is known as combustion, and in extreme cases results in an explosion which is a very rapid combustion. The natural fuels, wood, coal, oil, and gas, all contain to a greater or less extent the elements carbon and hydrogen in addition to other constituents. These two elements are responsible for practically all the heating value of the various fuels which are used, since they are capable, under suitable conditions, of being oxidized very rapidly and, in so doing, liberate large quantities of heat. Further, they unite with each other to form compounds known as hydrocarbons in a great variety of combinations, and these hydrocarbons form a very useful proportion of our fuel supplies.

Carbon itself is not easily ignited under ordinary conditions, nor is it easy to burn rapidly. Hydrocarbons, as a rule, ignite easily, and in a fuel containing carbon and hydrocarbon such as coal, the amount of hydrocarbon present decides the difficulty or otherwise of igniting the coal, and enables the fixed carbon in the coal to be satisfactorily burned.

In coal the hydrogen present is combined with some of the carbon, the remainder of the carbon being in its elementary form and known as fixed carbon. In liquid fuels the carbon and hydrogen both exist as hydrocarbons, several combinations being present. Petrol and paraffin are hydrocarbon fuels, but each contain fractions of several hydrocarbons and they are, therefore, not single substances. In gaseous fuels the hydrogen may exist as an element or in combination with carbon as a gaseous hydrocarbon. The carbon exists only in combination with the hydrogen as a gaseous hydrocarbon, or as carbon monoxide, which is a gas produced by the partial oxidation of carbon.

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Heat of Formation of a Compound. When two elements unite to form a definite compound, they do so in fixed proportions which cannot be varied so long as the same compound is formed. The combination may result in the liberation or absorption of heat, and the amount of heat thus liberated is known as the heat of formation of that quantity of the particular compound formed. In order to standardize the results the heat of formation is given with respect to a weight of the substance formed equal to the molecular weight in pounds or grammes, according to the unit of mass used. For example, when 12 lb. of carbon burn to carbon dioxide, the weight of carbon dioxide formed is 44 lb., and the heat liberated is 97,300 C.H.U. The heat of formation of carbon dioxide is thus said to be 97,300 C.H.U. If the unit of mass taken had been 1 grm. instead of 1 lb., the heat of formation would have been 97,300 calories. It is evident from this statement that if 44 lb. of carbon dioxide had to be decomposed into the elements of carbon and oxygen it would be necessary to supply 97,300 C.H.U. during the decomposition process, since this amount of heat had been liberated when the substance was formed.

When carbon and hydrogen unite to form a hydrocarbon, heat is liberated as a rule, but not invariably. In the formation of methane (CH_4), 21,700 C.H.U. are liberated for every 16 lb. of methane formed, whilst in the case of acetylene (C_2H_2), 47,800 C.H.U. are absorbed for every 26 lb. of acetylene formed. The decomposition of 16 lb. of methane into carbon and hydrogen would thus require the supply of 21,700 C.H.U., whilst the decomposition of 26 lb. of acetylene would be accompanied by the liberation of 47,800 C.H.U.

Calorific Values of Fuels. Since the heat of formation of carbon dioxide is 97,300 C.H.U., and 12 lb. of carbon are burned to dioxide to liberate this quantity of heat, it is evident that each pound of carbon burning to carbon dioxide liberates $\frac{97{,}300}{12} = 8{,}108$ C.H.U. The heat liberated during the combustion of 1 lb. of fuel is termed the calorific value of the fuel, and hence the calorific value of carbon burning to carbon dioxide is 8,108 C.H.U. Similarly, the higher calorific value of hydrogen is 34,200 C.H.U., and the lower calorific value 29,050 C.H.U. The meaning of the two values will be discussed later.

Calculations of Theoretical Calorific Value of a Fuel. If the analysis of a fuel be known, then from the amounts of carbon and hydrogen present per pound of fuel, and a knowledge of their calorific values, it is possible to calculate the theoretical calorific value of the fuel. To illustrate this, consider the theoretical calorific value of fuel oil. An average fuel oil will contain approximately 84 per cent of carbon and 12 per cent of hydrogen by weight, the remaining 4 per cent being sulphur, oxygen, nitrogen, and ash. Neglecting the heating value of the sulphur, which is relatively very small, the theoretical calorific value of the oil will be that due to the quantities of carbon and hydrogen in it.

Hence—

Heat liberated by the combustion of 0·84 lb. of carbon
$= 0{\cdot}84 \times 8{,}108 = 6{,}810$ C.H.U.
Heat liberated by the combustion of 0·12 lb. of hydrogen
$= 0{\cdot}12 \times 34{,}200 = 4{,}100$,,
∴ Calorific value of 1 lb. of fuel oil
$= 10{,}910$,,

This calorific value of 10,910 C.H.U. is appreciably in excess of the true calorific value for the following reason. If the carbon and hydrogen existed as elements in the fuel, then the theoretical calorific value would be the true value. They exist, however, as hydrocarbons in the fuel, and before they can liberate the heat indicated in the calculation these hydrocarbons must be decomposed into their elements. This decomposition necessitates the absorption of a certain amount of heat which was liberated on the formation of the compounds, and hence the true calorific value is less than that calculated by the amount of heat required to decompose the hydrocarbons in the fuel. Actually, with a fuel oil, the true calorific value is about 5 per cent less than the theoretical calorific value.

In practically all classes of solid and liquid fuels the necessity for decomposing one or other compound fuel element is present, and consequently the calorific value of such fuels calculated from the chemical analysis is erroneous, and this method cannot be used even where the analysis is known.

Various empirical formulae have been suggested for calculating the calorific value from a knowledge of the chemical analysis of specific fuels, but from what has been said it is evident that the results can only be very approximately correct. With gaseous fuels, where the fuel elements consist of hydrogen, carbon monoxide, and methane, it is possible to calculate the calorific value of the gas accurately, but if the gas contains some hydrocarbons whose chemical composition is not determined the method is not satisfactory.

Higher and Lower Calorific Value. When hydrogen or any fuel containing hydrogen is completely burned, some steam or water vapour is formed, and forms part of the products of combustion. So long as the temperature of the products is high enough this water vapour does not condense, since it is superheated, and the heat given out as the products cool is sensible heat only. As the cooling proceeds, however, a temperature is reached, called the dew point, where the water vapour present is dry and saturated, and further cooling results in progressive condensation of the water vapour, there being a definite amount of water vapour present uncondensed for any specific temperature, lower than the dew point. This condensation liberates latent heat and, in addition, the water of condensation in cooling gives up sensible heat. The sum of these two effects exceeds appreciably the amount of heat that would have been given up if the water vapour had remained as vapour during the cooling. In order to absorb all the heat given to the products by the combustion of any fuel, it is necessary to cool them down to the temperature of the air supply. When water vapour is present in the products it is evident that the amount of heat available when the products are cooled to atmospheric temperature is partly due to the condensation of the water vapour, and the calorific value obtained including this amount is called the higher, and excluding it the lower, calorific value. Since the difference between the two values is due to the presence of water vapour it is evident that any fuel containing hydrogen will have a higher and lower calorific value, whilst fuels which contain no hydrogen will have just the one calorific value.

Determination of Calorific Values. The obvious method of determining the calorific value of a fuel is to burn a known quantity, measure the heat given out due to the burning, and calculate the heat liberated per unit mass of the fuel, if it is either a solid or liquid, or per unit volume if it is a gas. This is the principle adopted in all types of fuel calorimeters, although the method of carrying out the principle differs with different instruments.

Before describing in detail the methods of carrying out the test or the apparatus used, it is as well to discuss the general problems involved and point out some of the difficulties which arise.

The heat liberated is measured by causing it to be absorbed by a known quantity of water, the water being supplied to the apparatus either in the form of a constant rate of flow or as a fixed quantity in the apparatus, and replaced for each test. Similarly, the fuel may be supplied continuously throughout the test and be burned at a constant rate, or it may be supplied as a fixed amount, previously determined by weighing. The constant rate of supply of water and fuel is used in practically all gas calorimeters, and the fixed quantities of fuel and water in calorimeters designed for testing solid and liquid fuels.

In order to cool the products down to the initial temperature of the air or oxygen supply the rise allowed in the temperature of the water is only small, so that accurate measurement of that temperature rise is essential, and it is furthermore necessary that the temperature measured should be truly that of the water.

To ensure the first, fine reading and accurately calibrated thermometers are used, graduated to either $\frac{1}{10}$° C. or $\frac{1}{100}$° C., according to the type of apparatus used, and a magnifying glass enables estimations to $\frac{1}{100}$° C. or $\frac{1}{1000}$° C. respectively to be made.

To be sure that the temperature reading is a true one, arrangements must be made to thoroughly mix the water before it reaches the thermometer, and this is done by passing the water through baffles in instruments using the constant flow supply, and by the use of stirrers in instruments using fixed amounts of water.

It is obviously essential that the quantity of fuel used should be measured accurately.

GAS CALORIMETERS

The general principles embodied in gas calorimeters may be best explained by reference to the diagrammatic sectional drawing (Fig. 1). In this calorimeter the water enters two annular tubes *B* and *C* at *A*, these tubes being connected at the bottom and top by a series of tubes *E* and *D*. Water entering at *A* passes upwards through *B* and *C* and, on leaving the calorimeter body, is thoroughly mixed by the baffles *F* before reaching the thermometer *G*. The water leaves the instrument at *H*, where it may be collected in a measuring flask or cylinder.

The products of combustion from the gaseous fuel pass upwards through the centre of the annular tube *C* and then downwards between *C* and *D* into the metal base *J*, and so out into the air. Condensation of steam from the products occurs on the walls of the tubes *B* and *C*, and the condensed water runs down the walls and falls into the metal base *J*, leaving in the form of a continuous drip at the outlet lip *K*. The whole apparatus is heat insulated to prevent radiation losses as much as possible by means of an annular air jacket *L*.

Method of Carrying Out a Test. The gas should be taken from a main through which gas is constantly passing. This can be arranged by having another burner continuously lit and supplied from the same main, and preferably farther away from the meter than the burner used with the calorimeter. The gas is adjusted until the rate of supply to the burner of the calorimeter is from 4 to 5 cub. ft. per hour with town's gas, and 16 to 20 cub. ft. per hour with producer gas. If producer gas is being tested a plain tube burner without any holes at the base is used, no additional air supply being necessary.

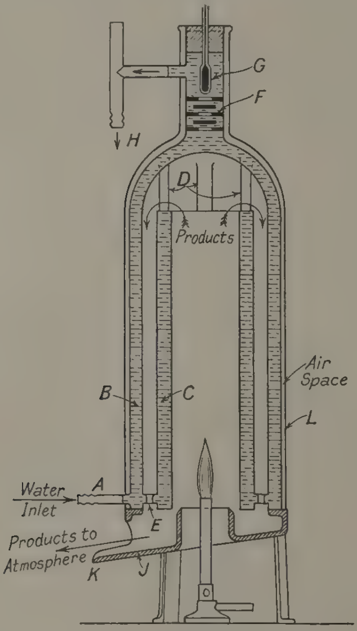

Fig. 1. Diagrammatic Section of a Gas Calorimeter

For gas measuring purposes a small precision gas meter (wet) is used with a large dial and pointer. One revolution of the pointer is equivalent to a tenth of a cubic foot, and the dial is divided into one hundred equal divisions, each representing $\frac{1}{1000}$ cub. ft. An inner dial graduated in cubic feet, and tenths of a cubic foot, is also provided. The pressure of the gas is measured in inches of water, a fairly sensitive water gauge being used, and if this pressure is not steady some form of pressure governor has to be used. The temperature of the gas in the meter is given by a suitably placed thermometer.

The water supply should come directly from the mains or from a large storage tank, holding sufficient water for a number of tests. For accurate work it is essential that the temperature of the inlet water should be constant, and if the main supply is used a tap should be left running for some time previous to the commencement of the test to enable a steady temperature of the inlet water to be attained. The accuracy of the test will depend on the constancy of the inlet temperature, and the rates of flow of water and gas to the calorimeter. The rate of water flow is adjusted until the rise in temperature of the water in passing through the calorimeter is approximately 15° C., although this value is not critical.

When these conditions have been obtained and remained steady for some time, the test is commenced by putting a measuring cylinder to collect the water discharged when the meter pointer passes the zero mark at the top of the fine reading dial, the reading on the gas meter being noted. As soon as this has been done readings of the outlet thermometer temperature are taken every 15 seconds, and of the inlet thermometer temperature every minute. These thermometers are graduated to $\frac{1}{10}$° C., and by means of a magnifying glass estimations to $\frac{1}{100}$° C. are made. When the water in the cylinder is approximately 2,000 cub. cm., the meter is again read when the pointer passes any convenient reading, and the water diverted from the measuring cylinder. The cylinder is taken away and the amount of water collected during the test noted. During the test the observer who is responsible for reading the meter and booking the records takes readings every minute of the pressure and temperature of the gas supply. The barometric height is obtained prior to the commencement of the test.

As a result of the test the following information is available—

Temperatures.
Mean inlet water temperature.
Mean outlet water temperature.
Mean gas temperature.

Pressures.
Mean gas pressure in inches of water.
Height of barometer.

Volumes.
Corresponding volumes of gas in cubic feet, and of water in cubic centimetres, used during test.

Example. To illustrate the method of working out the result of the test, the following data obtained when using a gas calorimeter are given, the fuel being town gas—

Quantity of gas used	0·28 cub. ft.
Duration of test	3 mins. 54 secs.
Rate of gas supply per hour	4·30 cub. ft.
Mean outlet water temperature	34·485° C.
Mean inlet water temperature	18·24° C.
Rise in temperature	16·245° C.
Volume of water collected	2,030 cub. cm.
Mean temperature of products leaving apparatus	69° F.
Mean air temperature	20·5° C.
Mean gas temperature in meter	22·0° C.
Mean gas pressure in meter	3·4 in. water
Height of barometer	29·78 in.
Condensed water	31·0 cub. cm.

collected during the burning of 2·10 cub. ft. of gas in 29·3 mins.

$$3{\cdot}4 \text{ in. of water} = \frac{3{\cdot}4}{13{\cdot}6}$$

$$= {\cdot}25 \text{ in. of mercury.}$$

Absolute pressure of gas supply

$$= 29{\cdot}78 + {\cdot}25 = 30{\cdot}03 \text{ in. of mercury.}$$

Temperature of gas in meter

$$= 22^\circ \text{ C.} = 71{\cdot}6\ ^\circ\text{F.}$$

Absolute temperature

$$= 461 + 71{\cdot}6 = 532{\cdot}6^\circ \text{ F.}$$

The standard conditions of temperature and pressure are 60° F. and 30 in. of mercury respectively, and the volume of gas used measured under the standard conditions would be

$$0{\cdot}28 \times \frac{30{\cdot}03}{30{\cdot}0} \times \frac{520}{532{\cdot}6} = {\cdot}2734 \text{ cub. ft.}$$

The weight of water collected during the test is 2,030 grm. or $\frac{2030}{453{\cdot}6}$ lb., and the temperature rise is 16·245° C. or 29·24° F.

Hence, heat absorbed by the water during the burning of 0·2734 cub. ft. of gas is $\frac{2030}{453{\cdot}6} \times 29{\cdot}24$ B.Th.U.'s, and gross calorific value of the gas in B.Th.U.'s per cubic foot

$$= \frac{2030 \times 29{\cdot}24}{453{\cdot}6 \times {\cdot}2734} = 478{\cdot}7.$$

This calorific value is the higher value, and if the lower or net value is required allowance has to be made for that portion of the heat given up by the products of combustion during cooling, which is due to the condensation of water vapour. In the test shown, 31·0 grm. of water were collected whilst 2·10 cub. ft. of gas measured under meter conditions of temperature and pressure were burned. Under standard conditions of temperature and pressure this volume would be

$$2{\cdot}10 \times \frac{30{\cdot}03}{30{\cdot}0} \times \frac{520}{532{\cdot}6} = 2{\cdot}05 \text{ cub. ft.}$$

and consequently the condensation per cubic foot of gas would be $\frac{31{\cdot}0}{2{\cdot}05}$ grm., or $\frac{31{\cdot}0}{2{\cdot}05} \times 453{\cdot}6$, or 0·0333 lb.

It is customary to assume that each pound of condensate is equivalent to 970 B.Th.U.'s, which is the latent heat of vaporization at atmospheric pressure, and using this conventional figure, the heat due to the condensation of 0·0333 lb. would be $970 \times 0{\cdot}0333 = 32{\cdot}3$ B.Th.U.'s.

The net or lower calorific value of the gas would thus be

$$478{\cdot}7 - 32{\cdot}3 = 456{\cdot}4 \text{ B.Th.U.'s.}$$

The method of testing coal gas, and the apparatus to be employed to comply with the demands of the Gas Regulation Act, 1920, are laid down in a booklet published by His Majesty's Stationery Office, entitled *General Notification of the Gas Referees*, and readers interested should make themselves familiar with the methods and precautions there outlined.

For accurate results considerable care is necessary, and one or two corrections have to be made which have, for the sake of simplicity, been omitted in the example worked out.

STEAM BOILERS AND THEIR ACCESSORIES

By Percy E. Rycroft, M.B.E., M.I.E.E.

LESSON XI

OIL FIRING, WASTE HEAT, REFUSE

Although in this country the relative cost of fuel oil and coal is usually much in favour of the latter, circumstances sometimes arise in which it is advantageous to install oil firing.

Assuming that a good average residual oil has a calorific value of 18,500 B.Th.U.'s against 11,000 B.Th.U.'s for a good bituminous slack, and making due allowance for savings on fuel handling, reduced labour costs due to the absence of ashes, and other economies, it may be taken that it is a commercial proposition to fire with oil, providing the cost of the oil fuel is not more than twice as much as that of the coal.

If this proportion is exceeded there must be other and weighty considerations to warrant its adoption.

There are two main systems of oil burning, viz., steam atomizing and pressure atomizing, with a possible third alternative of substituting compressed air for steam under the first heading.

Steam Operated Burners. The steam atomizing system requires only a simple equipment consisting of oil and steam supply pipes connected to a common burner, the oil flowing down the centre of the burner to a small orifice and being met just before emergence by a jet of steam which breaks it up into a fine spray. In some burners the steam is arranged to impart a rotary motion to the oil spray as it enters the furnace, and this shows some advantage over an ordinary flat jet of steam.

For operating purposes the oil fuel is heated—usually by means of a steam coil in the oil service tank—in order that it may flow freely.

Steam burners can be constructed to deliver up to about 600 lb. of fuel oil per hour, but a normal working load may be taken at about 250 lb. per burner per hour with a damper draught of about ½ in. water gauge.

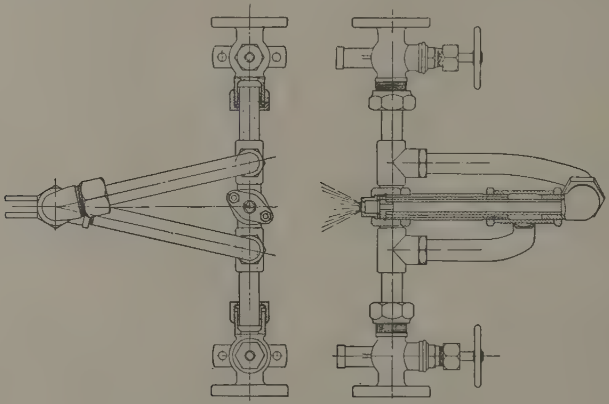

Fig. 69. Oil Fuel Burner, Using Steam as the Atomizing Agent

From 2 to 5 per cent of the total steam produced by the boiler is required for the burners.

Pressure Systems. The pressure system of oil fuel firing is appreciably more efficient than that of steam operated burners, the complete equipment consisting of oil heater, filters, pressure pump, pipe lines, and burners.

The oil is heated to ensure a steady flow of fuel, and filtered to prevent any small particles of foreign matter from blocking the very small holes in the burner orifice plate.

As a further safe-guard against this possibility, arrangements are usually made so that the actual burner can be quickly removed for cleaning or replacement without interfering in any way with the operation of adjacent burners.

The oil pumps are usually of the steam-driven double-acting type, and a normal working pressure is about 145 lb. per sq. in. Pressure operated burners can be constructed for a duty up to about 900 to 1,000 lb. of fuel per burner per hour, but are usually operated at about 250–400 lb. per hour.

Air for combustion is provided by means of registers which surround the burner, and these can be regulated as may be necessary.

Duplicate pumping, heating, and filtering plant should always be installed and, in applying any system of oil firing to steam boilers, it is particularly important to arrange the burners so that there is no continuous impingement of the flame jet on the water surfaces. It is sometimes necessary to protect by firebrick the boiler parts at all liable to such impingement, while similar and adequate protection should be given to the ash-pit, particularly when any steel girders or framing form part of the lower structure of the furnace.

Waste Heat. Under the heading of waste heat we include furnace gases from blast furnaces, coke ovens, cement kilns, steel works, gasworks, and similar industrial plants.

These works consume large quantities of fuel in the manufacture of their main products, and have very considerable supplies of surplus heat available which would otherwise be lost unless used for steam raising purposes.

In the case of coke ovens, puddling furnaces, reheating and reverberatory furnaces and the like, relatively high gas temperatures—up to as much as 2,200° F.—are often available.

The draught necessary for combustion with these plants is not high and, therefore, the introduction of a steam boiler in the circuit offers little difficulty as there is not material interference with the furnace draught, due to the lowering of the chimney exit gas temperature.

With regenerative and other furnaces, however, the initial gas temperature available for steam raising will probably not exceed 1,200–1,300° F., and may be as low as 900° F., so that apart from the question of maintaining a suitable draught at the furnace outlet it is necessary to increase the velocity of the relatively cool gases through the boiler in order to obtain a reasonable heat transference.

This is readily done by the introduction of mechanical draught, and it is advisable to remember that the supply of gas is often irregular, and that ample margin should be allowed in the size of the induced draught fans, so that they can readily deal with increased outputs from the furnaces above ordinary normal conditions of working.

In the case of coke ovens, there is often a surplus of "live" gas available in addition to actual waste heat. This live gas can be readily fed into a boiler furnace through a Bunsen or multiple type burners and burnt in conjunction with the waste gases.

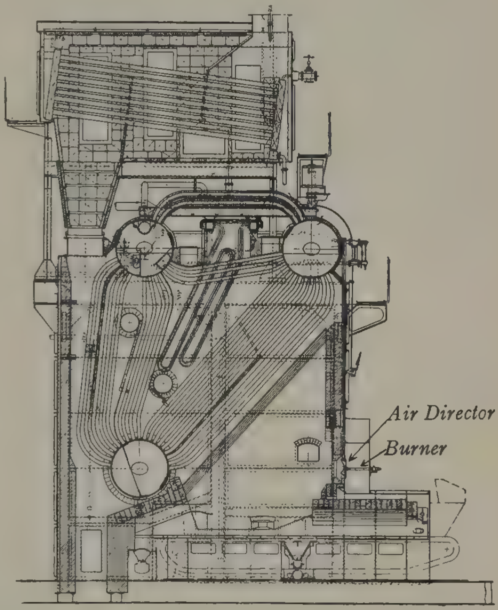

Fig. 74. Section of Stirling Boiler Equipped for Burning Oil as well as Coal

With cement kilns the problem of utilizing the waste heat is a little complicated by the fact that the gases leaving the kilns are heavily laden with dust. There is, however, very little tendency for this dust actually to stick to the boiler tubes and surfaces, and it can be readily kept clean by means of steam operated soot blowers.

As in the case of regenerative furnaces it is essential that the draught at the kiln mouth should not be reduced, and induced draught fans are again employed.

The volume of gases to be dealt with is large, and care is necessary to reduce air leakages to a minimum in order to avoid overloading the fans by infiltration of this kind. It is usually an advantage in this type of plant for all flues and boiler brickwork to be steel cased.

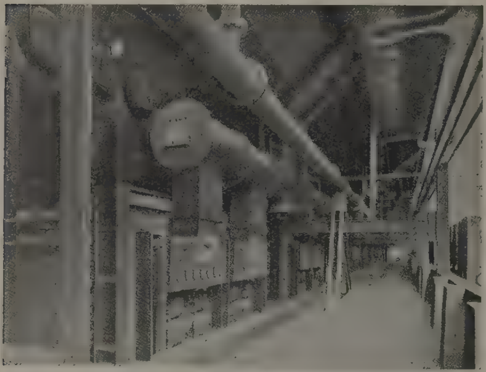

Fig. 75. View of a Typical Boiler Installation Fired with Blast Furnace Gas—Babcock Boilers

Waste gases are always liable to be charged with a certain percentage of sulphur, and this taken in conjunction with any steam leaks or other moisture is liable to set up corrosion, particularly when the final exit temperatures are low enough to allow condensation on the relatively cold boiler surfaces; but reasonable precautions will prevent this particular trouble.

The temperature of the gases available from different process plants will, of course, vary appreciably, and this has to be considered when determining the heating surface required for any particular layout, but beyond this and the question of gas velocity, which has already been referred to, no very special modifications to a standard steam plant are involved in order to use waste heat.

Refuse Burning. There are many industries which in the course of their process work provide an appreciable amount of waste material, which contains a certain amount of combustible matter that can be used under steam boilers.

Among such fuel may be mentioned spent tan, sawdust, and wood chips, coffee husks, bagasse, sugar cane, and many others.

In addition to these waste products there are many thousands of tons of house refuse to be disposed of daily, and refuse destructors have been established in many parts of the country for this purpose.

The burning of these waste fuels offers no particular difficulty except that some modification to the furnace is usually required; a "stepped" or inclined hand fired grate is often installed and a forced draught equipment is generally necessary.

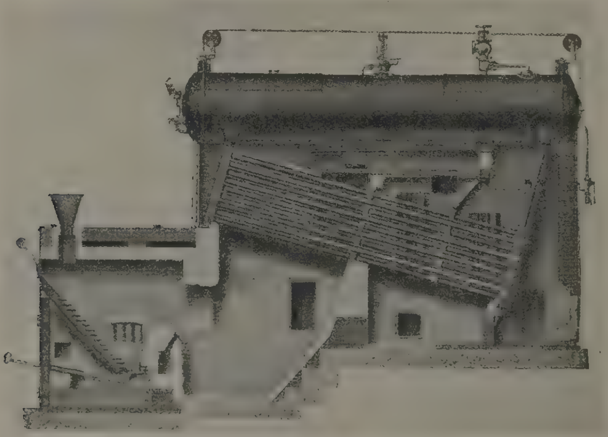

Fig. 76. Cross-section of Furnace Equipped with Stepped Grate for Burning Spent Tan and Similar Refuse

The boiler construction is not materially affected except that the furnace temperature with these waste fuels is invariably less than with coal or oil fired plants, and a lower rate of evaporation per square foot of heating surface is therefore allowed for.

Most waste fuels contain a high percentage of moisture—often over 40 per cent—and under these conditions some trouble is likely to be experienced in maintaining ignition. To deal with this problem, " drying hearths " are sometimes installed, the fuel being banked on to a firebrick slope at the front or, where a top feed is adopted, rear of the furnace, so that a good deal of the moisture can be driven off before the fuel is allowed to enter the furnace proper.

Although in some cases house refuse is fired directly on to grates immediately under the boiler, it is more usual for this material to be burnt in a number of furnaces arranged in cellular form, the gases from which pass into a common flue and from thence to the boilers.

The calorific value of house refuse rarely exceeds 4,000 B.Th.U.'s per lb., and the furnace temperatures are relatively low, 1,600° F. being a fair figure.

An evaporation of 1 lb. of water per lb. of refuse consumed is a quite reasonable performance, although higher claims than this are sometimes made.

Mechanical forced draught or steam blowers are essential for refuse burning.

POWER TRANSMISSION BY CHAINS

By H. M. Andrew, Wh.Ex.

LESSON IV

THE SELECTION AND USE OF CHAIN DRIVE

SELECTION

The objects aimed at in the selection of a power transmission generally consist of one or more of the following five desirable qualities. While it is obviously advantageous to obtain all these qualities to their maximum in a drive, this is generally not practicable and, consequently, it is usually necessary to decide which of these qualities we need most particularly for our purpose.

1. **Long Life.** Generally the foremost requirement in industrial drives, such as lineshafts, etc. The life of a drive is dependent partly on the amount of bearing pressure in pounds per square inch. It is still more dependent on the success with which flapping and irregular driving is avoided. It is most dependent of all on the provision of suitable protection from dirt and adequate lubrication, a requirement which will be dealt with later on in this lesson.

2. **Low Initial Cost.** An object to be always aimed at, with due regard to the other requirements. The initial cost depends largely on the type and pitch of chain used. The smaller the pitch—down to, say, about $\frac{1}{2}$ in. pitch—the cheaper the chain. Saving in first cost on the protection and proper lubrication of the drive is generally false economy.

3. **Quietness.** An essential in motor-car drives and in many industrial drives. Quietness is best obtained by the use of inverted tooth chain where this is practicable, but a fair degree of silence can be obtained with roller chain, given ample lubrication, on wheels with large numbers of teeth. In either type, large numbers of teeth on the wheels are always very advantageous in securing quietness.

4. **Lightness.** Often the first essential in such drives as aeroplane propellers. Light weight is best obtained by the use of bush roller chain, and the smallest pitch that provides sufficient strength and bearing area for the work.

5. **Robustness.** For cases where shock load is present in some pronounced degree, bush roller chain is more resistant to shock and fatigue than inverted tooth chain. In drives up to, say, 500 r.p.m., this quality can be obtained to any required degree by increasing the pitch and therefore the strength of the chain, but in high speed drives this is not always a

successful method as the increased weight of the chain may increase centrifugal tension and the effect of shock proportionately.

The usual information available for selection is the horse-power to be transmitted, the revolutions per minute of the shafts, and the space available for wheels. The type of chain to be employed is chosen with due regard to the qualities required of it.

At the maker's limit of, say, 2,000 lb. per sq. in. bearing pressure, *maximum* horse-power from any pitch of bush roller chain would be obtained at about 3,700 ft. per min. speed, or in the case of inverted tooth chain at about 2,400 ft. per min. The most economical speed to run at is about 2,500 ft. per min. for bush roller, and 1,700 ft. per min. for inverted tooth chain. While 1,700 ft. per min. is quite a usual speed for inverted chain, such a speed as 2,500 ft. for bush roller chain can only be employed with reasonable quietness by using large numbers of teeth in the wheels and heavy lubrication.

The actual limit to speed in the case under consideration, expressed in feet per minute, equals the revolutions per minute of a shaft multiplied by the circumference in feet of the available space for the wheel. If, as is usually the case, this available speed is much less than the speeds discussed above, it indicates that the whole space available should be utilized so that the wheels may have the maximum possible number of teeth.

A suitable speed to run at can be determined in the above manner, and this speed $S = \frac{RNP}{12}$ in which S and R are known, while NP equals the pitch circumference (approx.) of the chain wheel.

T (the tension due to horse-power)

$$= \frac{H \times 33{,}000}{S}$$

C (the centrifugal tension) depends on the weight of the chain used, and equals $\frac{WS^2}{115{,}920}$

$\frac{T + C}{A}$ = the bearing pressure in pounds per square inch, which should not exceed, say, 2,000 lb.

$\frac{\text{Strength of chain}}{T + C}$ = factor of safety, which should in the average case be about 15 for bush roller and 20 for inverted tooth chain.

If shock load is present in the drive, the safeguard is a high factor of safety.

The chain makers' catalogues generally contain selection tables that are a useful guide to work with. The main object in selection should be to obtain as many teeth as is practicable in the chain wheels, as this makes for good efficiency, quiet and smooth running, and minimum wear. This effect is obtained by selecting the smallest pitch of chain that will safely carry the load. The chain should have a minimum angle of lap around the driving wheel of about 120° to ensure a good grip of the chain by the wheel, and the driven wheel or wheels should have not less than 90° preferably. Adjusting jockey wheels need only have a few teeth in mesh, according to the degree to which correct adjustment is going to be maintained.

LUBRICATION

The great importance of efficient lubrication has been previously emphasized, and a few notes on methods of lubrication may be of use in indicating how best to obtain good results.

The bearing surfaces of a chain are not subject to continuous pressure, but have an intermittent load varying rapidly from zero to maximum. The best method of lubrication is therefore, as far as possible, to seal the bearings all round with a film of lubricant so that this lubricant will be forced into the bearing when the pressure is relieved. The best medium for the purpose is thin mineral oil applied fairly copiously to the *inside* face of the chain, so that centrifugal action will tend to throw it through the chain. Inverted tooth chains are easy to lubricate in this manner, but in bush roller chains the lubricant has to pass down between the inner and outer link plates in order to reach the pin bearing. Obviously, tallow or graphite grease is unsuitable if simply pasted on to the chain. If it is required to use solid lubricants of this type the chain should be soaked in the heated lubricant for an hour or so until some air in the bearings has been expanded out and replaced by a film of graphite or grease. The chain will then be good for a limited period of service (according to the nature of its duty) and should then be retreated. Such lubricants are really only suitable for slow speed drives. For high speed drives the most suitable method to use is a copious supply of pure mineral oil pumped on to the inner face of the chain. Radial throw of oil through the rim of the wheels, as mentioned in Lesson I, is also effective.

Running in an oil bath is quite efficient provided the level of oil is maintained sufficient to fully immerse the lower part of the chain.

The amount of protection to be afforded to the chain by a guard or casing depends on whether dirt and grit are liable to get into the drive. Foreign matter should always be thoroughly excluded to prevent it forming a grinding compound with the oil; and where copious lubrication is used a casing is essential to conserve the oil, which should preferably be filtered through gauze before re-use. In large industrial drives a sight gauge should be incorporated to give a visible indication that oil is present in the case to a suitable amount.

Adjustment. Just as a belt or rope is shortened to take up stretch so can a chain be shortened to take up wear by removing one or more pitches. When an odd number of pitches is required, the odd link is termed a cranked link, or a hunting link, as it has to join up an inside combination to an outside combination. Odd pitches of this kind with bent links are best avoided in high speed or heavy duty drives, as they can only be made of equal strength to the rest of the chain by increasing their relative weight, and this slightly disturbs the smooth balance of the chain. For the same reason, high speed drives are often riveted up as an endless chain to avoid the use of joint links. The removal of a pitch of the chain naturally provides only a coarse method of adjustment, and if the chain is to be kept in close adjustment, which is always desirable in any drive, and really essential in high speed drives, either the centre distance of the wheels must be increased (as can be done by moving a motor on its slide rails) or an adjustable jockey wheel must be fitted to the slack side of the chain. A jockey is sometimes useful on the exterior face of a chain drive if it is desired to increase the angle of lap of the chain on either of the wheels.

Judgment must always be exercised when adjusting a worn chain, and sufficient slack must always be left in a stationary chain to allow it to ride out on the wheels as described in Lesson I. If a worn chain is drawn taut by its adjusting device it is liable to malgear when running. In the case of large heavy chains with long horizontal centre distance between wheels the slackness of the chain must be sufficient to allow for enough catenary sag to prevent any excess pressure on the chain bearings or the shafts supporting the wheels. The following points are a rough guide to adjustment—

1. In horizontal slow speed heavy drives, leave ample slack, increasing with centre distance.
2. In high speed heavy drives use a rather closer adjustment, particularly if any cyclic inequality is present in the drive. Increase the closeness of the adjustment according to the angle of the drive from the horizontal.
3. In vertical drives, slow or fast, heavy or light, use close adjustment until a slight beginning of tension can just be felt in the chain bearings.
4. For slow speed light chains maintain fairly close adjustment if cyclic irregularity is present, otherwise an ample slack can be allowed (although it is not necessary).
5. For high speed light drives maintain a close adjustment, particularly if cyclic irregularity is present or if the driving side is on the bottom of a nearly horizontal drive.

THE MANUFACTURE OF WIRE

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Records show the making of wire to be one of the most ancient of the metal-working crafts. It no doubt came into being when man first required a piece of drawn out metal to make either weapons, tools, personal ornaments, hooks, or other form of fastening. The first piece of wire ever made was most likely formed out of copper or gold, as both of these metals would be found in their native state, their plastic properties enabling the workmen to beat out the metal in the easiest possible manner.

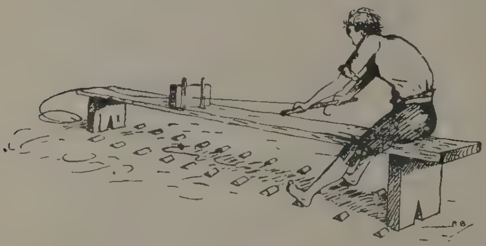


Fig. 1

When iron came into use, rod or wire would, of course, be forged directly from a lump of crude iron; afterwards the time came when iron was beaten into thin plates, cut into strips, and rounded by hammering or rubbing. Then later there followed the method of making wire by drawing the metal through a conical hole in a hard plate.

The early metal workers do not seem to have been acquainted with the process of making wire by passing the rough rod through a draw-plate, for as long as the wire was formed by the hammer the metal workers by whom it was made were styled wire-smiths, but with the introduction of the drawing process their designation was changed to wire-drawers or wire-millers. There is fairly definite information that wire was drawn in this way about the thirteenth century, but as the process is so simple it is quite possible that much earlier metal workers used this means of reducing the diameter of a piece of wire or rod.

The earliest wire made by the use of the draw-plate was manufactured by main force, the workmen pulling the wire directly through the hole in the draw-plate, as shown in Fig. 1. It was then an easy stage to the application of power, the first kind used being water power. One of the last of the old mills of this character is that at Tintern Abbey.

Before the introduction of grooved rolls about 150 years ago, the iron rods which form the raw material for wire-drawing were manufactured by hammering lumps of crude iron into rods of about the thickness of the little finger. They were then run down or reduced by a method of coarse drawing called "ripping" or "rumpling," and after this were carried on through smaller and smaller dies until the required size of wire was obtained.

Modern Methods. When the rolling of bars came into use, wrought iron for wire-drawing purposes was rolled down into rods of about $\frac{1}{4}$ in. diameter, the weight of the piece usually being about 28 lb. The introduction of mild steel or ingot iron has changed the character of the wire-drawing trade, for, as in other industries, mild steel has almost pushed the use of wrought iron out of existence. With the exception of a little Swedish charcoal iron, wrought iron has almost ceased to be used in the making of wire. Mild steel is usually purer and more uniform in its composition than ordinary wrought iron, and consequently much heavier pieces can now be drawn than obtained in the old days.

Whilst the raw product that comes into a wire works is the rolled rod from the steel mill, it is, perhaps, as well for us to look back a little at the earlier stages through which the steel has to pass before it comes to the wire mill. The steel, of course, originally comes from the iron ore which is smelted in the blast furnace. After smelting, the hot metal passes to the steel furnace and is there purified. In this process its impurities—such as phosphorus, silicon, sulphur, etc.—are very largely removed. After being stripped from the mould the ingot is reheated or soaked, passed through the cogging mill, and ultimately rolled down into billets of about 2 in. square.

To produce the rods the billets are charged into a furnace and heated up to the required temperature, afterwards being passed through a continuous train of rolls and reduced down to about one-fifth of an inch in diameter.

The Cleaning of Rods and Wire. One of the most important processes in connection with the making of wire is the proper cleaning of rods, so as to free them entirely from any form of scale, as this latter is fatal to good wire-drawing. The general practice is to submerge the rods in cisterns of dilute hydrochloric acid until the whole of the scale has been completely "pickled" away. After cleaning in this form the rods are then washed well and allowed to stand until they are "browned" or coated with a film of ferric-hydrate. This "coating" of wire is exceedingly important and should be carried out with great care, as the subsequent drawing of the wire, especially for what is known as long-holing or carrying the wire to fine gauges, depends very largely upon the proper coating of the wire.

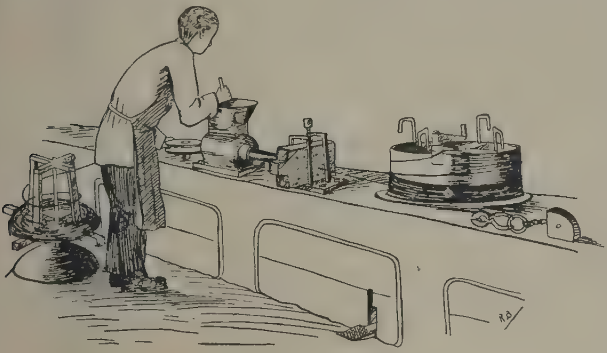


FIG. 2

After coating, the rods are dipped into a vat of hot lime water, lifted out, and well dried in ovens. The thin coating of lime on the top of the ferric-hydrate serves the double purpose of protecting the brown coat, and also of keeping the rod surface from corroding.

Before either rods or wire can be passed through the draw-plate it is necessary that the ends should be pointed to a conical form; this was originally done by hammering or filing, or a combination of both, but the bulk of the pointing at the present time is done by the aid of either rotary swaging machines or grooved rollers, and, in some cases, other special mechanical devices are used for forming the point.

Methods of Wire-drawing. There are two general processes of wire-drawing in vogue—one known as the "dry," and the other as the "wet" method. In dry drawing the rod or wire is passed through a draw-plate or die which has a tapered hole, the lubricant in this case being a special form of well dried soap. This method of drawing is shown in Fig. 2, in which the operator is seen ramming the soap down on to the rod as it passes through the soap box on its way to the steel die. The illustration shows the draw bench, the operation being known as single-holding, i.e. as the crude rod passes from the "swift" or cradle shown on the left hand and is ultimately wound on to the block on the right it is reduced one hole, which may be from a 5 gauge rod down to 8 gauge wire—the time of this operation for a 160 lb. piece being about five minutes. There are various ways in which the blocks are operated, sometimes ten or more being driven by bevel gearing from a line shaft, or they may be in groups of two or three, or in some cases driven directly with a motor connected to the vertical spindle.

In wet drawing the wire or rod is first coated with a thin film of copper, the latter being deposited by passing the rod or wire through a special solution containing sulphate of copper. The wire then passes through a soapy solution. and on through the hole of the draw-plate. This method of wire-drawing a hard metal within a shell or skin of a softer metal has been in use for more than a century, and in practice gives very good results.

The rods as they come into the wire mill are usually in pieces weighing about 150 to 170 lb., which will give a length of about a quarter of a mile of 5 gauge. This, when drawn to 10 gauge, increases in length to about three-quarters of a mile. If drawn to 20 gauge its length works out to about nine miles, and then if drawn further to 30 gauge stretches to a length of 70 miles, or 280 times the length of the original rod. Sometimes rods of twice the above weight are used, these, of course, giving wire of double the length stated.

A continuous process of wire-drawing is now in common use. The method consists in carrying the wire continuously from one block to another through interposed draw-plates, and thus on to a final winding block. The advantage of this method is that it obviates the handling of the wire in between the various passes or drafts.

Science of Wire-drawing. Whilst the casual observer would think that the making of wire is a very simple process which involves little knowledge or skill, in actual practice to obtain the best results a great amount of consideration has to be bestowed on the kind of dies used, the shape of hole, speed of drawing, method of lubrication, and many other points that affect not only the drawing of the wire, but its ultimate quality.

The reducing of the diameter of the metal rod by pulling it through a conical hole of smaller diameter is in its essence entirely different to either swaging or rolling, as it is the actual pull on the wire that sets up the pressure on the reducing shoulder which causes the reduction in the diameter of the rod or wire. The real action is of a duplex character, for at the same instant as the pressure is set up on the shoulder the metal at the core of the wire is made to slightly flow in advance of the skin, this causing the continuous collapse of the material which allows it to continuously pass forward through the die.

Careful experiment and calculation show that the steel in passing through the wire-drawers' plate is subjected to an enormous pressure, this in some cases amounting to as much as 150 tons to the square inch. With such great pressure on the shoulder of the tapered hole in the draw-plate it is manifest that a solid or semi-solid lubricant must be used in wire-drawing. The commonest lubricant employed is the best hard, dry olive oil soap.

It is essential in wire-drawing that there should never be metallic contact between the surface of the wire and the surface of the hole in the draw-plate, as this causes the wire to scrape and wear out the hole in the plate, producing unsizable wire. The arrangements for lubricating must be such that a film of the hard soap travels on the surface of the wire, and keeps it from coming into contact with the surface of the hole in the plate.

The speed at which wire travels through a draw-plate varies according to the diameter and quality of the material to be produced. For soft steel this speed may run up to more than 1,000 ft. per minute. The reduction of area per draft varies according to the quality of steel being drawn and the kind of wire required. It may run up to 40 per cent, and down to as low as 10 per cent.

The physical properties of steel are altered as it passes through the wire-drawing process, the chief alteration being a rapid increase in its tensile strength, and a reduction in its elongation. A No. 5 gauge rod (·212 in.) of ·14 per cent carbon steel, as it comes from the rolling mill, will have a tensile strength of about 33 tons per square inch. In drawing down to 16½ gauge its tensile strength rises to about 62 tons per square inch. This again after annealing, when drawn down to 33 gauge, mounts up to about 85 tons per square inch. A ·75 per cent carbon steel may start with a 5 gauge untempered rod, having a tensile strength of about 57 tons per square inch, but when drawn down to 33 gauge, with two intermediate temperings, it runs up to the extraordinary strength of 200 tons per square inch.

When stress-strain diagrams for mild steel wire are taken on an automatic recording machine, the chief characteristic observed is the great reduction in the elongation as the wire is reduced in diameter. Similar curves for hard steel wire indicate that whilst the untempered rod shows a definite yield point, the tempered rod and the drawn wire give no clearly defined yield point.

The limit of elasticity for both mild steel and hard steel wire is not easy to determine on account of the very slight deviation from the law of proportionality which obtains with wire under a small load.

The operation of wire-drawing on steel, it might be mentioned, has the effect of increasing its electrical resistance, so that in practice it is necessary to anneal wire that is to be used for telegraph and similar purposes.

The various physical properties that are usually mentioned in connection with wire are: its tensile strength, elongation, limit of elasticity, torsion and bending.

The softer metals such as copper, and alloys such as brass, are drawn in a similar manner to steel, the difference being in the methods of cleaning and annealing, also with the softer metals a larger use is made of methods of continuous drawing, in which wires may be run down to very fine gauges by the simultaneous reduction of nine or more passes in one operation.

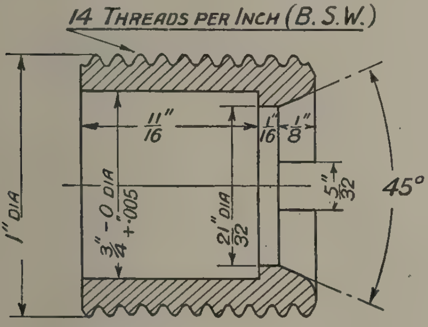
MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XXVIII

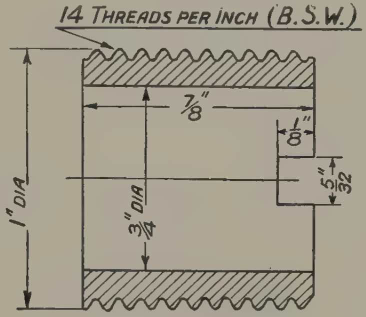
British Standards. We have referred to the place of the standard drawing as a reference for machine parts in comparatively frequent demand. Such drawings are often made to suit a single manufacturer's convenience in production. We have yet to consider the place of the standard drawing having a universal recognition.

Certain machine part proportions, such as the Whitworth thread for nuts and bolts, have already acquired this general notice, but there are other common details which have not yet obtained a definitely recognized place in the list of British Standards.



Standard Screwed Gland For ¾" Ext Dia Condenser Tube

Fig. 1



Screwed Gland For ¾" Ext Dia Condenser Tube

Fig. 2

Mention has already been made to the B.E.S.A. and its work (Sir Archibald Denny, Bart., LL.D., page 729, and Arthur P. Haslam, M.I.E.E., page 334), and we may now consider the drawing office attitude to the publications issued by the above association.

First, we may observe that, in the interest of utility alone, a designer ought always to endeavour to adhere to the proposed standard specification or drawing, and so help to obtain for all proposals the definite recognition accorded to such standards as nuts, bolts, and wire gauges.

Such observance of the proposed standard is sometimes frustrated by the demands of economy or requirements peculiar to the service required of the detail in question.

Customers' Preference. Further, it often happens that the prospective owner of the machine has ideas regarding the design which fail to agree with the standard specification, and in the interests of good business it is advisable to respect the demand at the expense of ignoring the standard. Such demands are not necessarily fads, but are often the result of experience which may be peculiar to the service required of the machine in question.

In such cases it is the duty of the designer to essay a successful compromise by satisfying the owners with the least departure from the standard.

Departure for Economy. Fig. 1 shows a detail of the B.E.S.A. standard screwed gland for a ¾ in. external diameter condenser tube. In the case of the glands fitted to the water inlet ends of the tubes the bell mouth tends to prevent the formation of a *vena contracta*, and yet some engineers will sacrifice this feature of efficiency to the claims of economy, specifying the much cheaper gland shown in Fig. 2.

Where owners show a preference for a design differing from the standard it is not advisable to unduly press the claims of the standard design. As an alternative, we may suggest a design modified in the less essential features. For example, the most important feature of the standard design shown in Fig. 1 is the screwing, because this affects the space for gland packing, interchangeability, and ease in replacing defective glands. This observance of the essential has been kept in the case of the cheaper design shown in Fig. 2, and may be adopted without prejudice to future fitting.

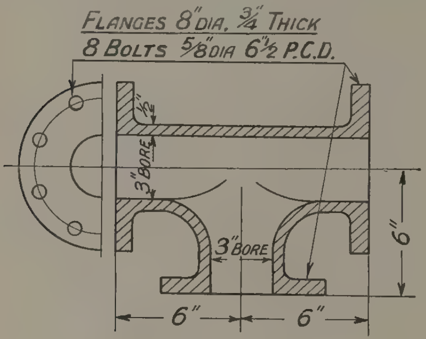


FIG. 3

Departure to Suit Existing Patterns. Fig. 3 shows a 3 in. bore cast-iron tee piece for a steam pressure of 125 lb.-sq. in., designed to suit the B.E.S.A. standard for such fittings. Unfortunately, the standard dimensions for such fittings has not always received strict observance with the result that many existing patterns do not agree with the standard.

Where an existing pattern is available we may observe economy by making the least alteration to the pattern and departing from the standard in the less important details. The more essential features of the standard design are interchangeability and the prompt replacement of defective parts.

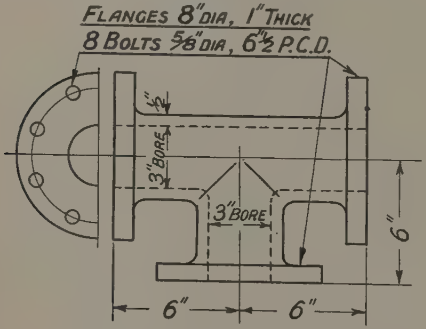


FIG. 4

Suppose a 3 in. bore cast-iron tee piece for 120 lb.-sq. in. pressure is required for an oil system, where leakage would be dangerous. We may, in order to guard against the possibility of leakage, increase the flange thickness above the standard dimension. Also, assuming there is a fairly similar pattern available, we may depart from the standard in the less important points and design the tee piece shown in Fig. 4.

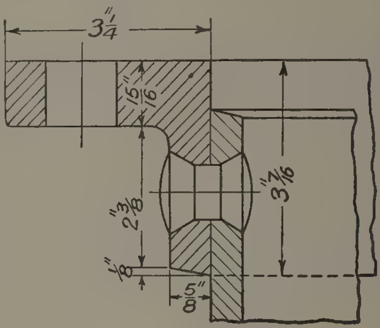


FIG. 5

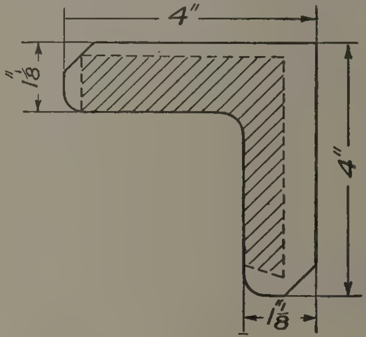
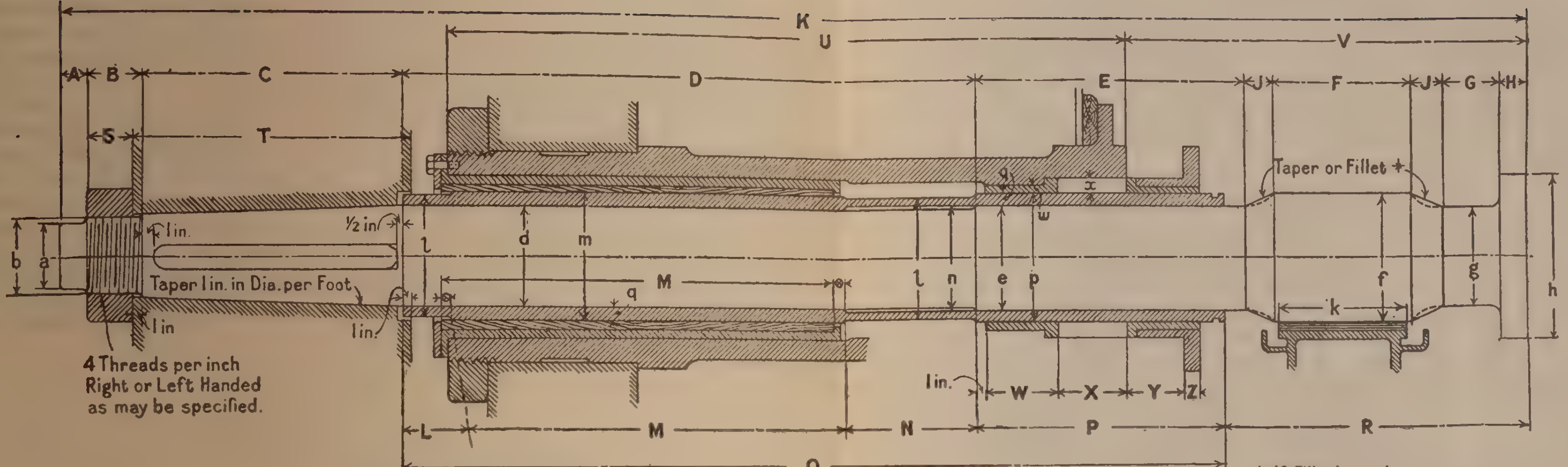


FIG. 6

Special Rolled Steel Section. Fig. 5 shows a sectional detail of a flanged joint for a large exhaust steam fitting, and it is desired that the section be made from a B.E.S.A. rolled steel angle section. Referring to the standard dimensions for rolled steel sections we observe the nearest section to the required joint is a 4 in. ×



4 Threads per inch
Right or Left Handed
as may be specified.

⊗ This dimension to be same as thickness of flange of bush.

* If Fillet is used radius to be 2J-1 inch.

Shaft.																	Bearing.	Liner.										R	Lengths.			Stern Tube.		Stuffing Box.						Diam. of Shaft.
Diameter.							Lengths.											Diameters.				Thickness.	Lengths.						Nut.	Boss.	Bush.									
d	a	b	e	f	g	h	A	B	C	D	E	F	G	H	J	K	k	l	m	n	p	q	L	M	N	P	Q	R	S	T	M	U	V	w	x	W	X	Y	Z	d
ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.
6	3¼	4	6⅛	7¼	6	13	2	4½	17	34½	20	10	3¼	1¾	2½	98	8	6⅞	7	6⅛	7⅛	½	5	25	4½	18½	53	21½	3½	19	25	42	27½	5/16	1	4½	7	3½	1⅛	6
6¼	3½	4¼	6⅜	7½	6⅛	13	2	4½	17	34½	20	10	3¼	1¾	2½	98	8	7⅛	7¼	6⅜	7⅜	½	5	26	3½	18½	53	21½	3½	19	26	42	27½	5/16	1	4½	7	3½	1⅛	6¼
6½	3¾	4½	6⅝	7¾	6⅜	13¾	2	4½	17	34½	20	10	3¼	1¾	2½	98	8	7⅜	7½	6⅝	7⅝	½	5	27	2½	18½	53	21½	3½	19	27	42	27½	5/16	1	4½	7	3½	1⅛	6½
6¾	4	4¾	6⅞	8	6⅝	13¾	2	4½	17	34½	20	10	3¼	1¾	2½	98	8	7⅝	7¾	6⅞	7⅞	½	5	28	1½	18½	53	21½	3½	19	28	42	27½	5/16	1	4½	7	3½	1⅛	6¾
7	4¼	5	7⅛	8¼	6⅞	14	2	5	19	39½	20	11	3¾	1¾	2½	107	9	7 15/16	8 1/16	7⅛	8 3/16	17/32	5	29	5½	18½	58	23	4	21	29	47	29	5/16	1	4½	7	3½	1⅛	7
7¼	4¼	5	7⅜	8½	7⅛	14	2	5	19	39½	20	11	3¾	1¾	2½	107	9	8 3/16	8 5/16	7⅜	8 7/16	17/32	5	30	4½	18½	58	23	4	21	30	47	29	5/16	1	4½	7	3½	1⅛	7¼
7½	4¾	5½	7⅝	8¾	7⅜	14¾	2	5	19	39½	20	11	3¾	1¾	2½	107	9	8 7/16	8 9/16	7⅝	8 11/16	17/32	5	31	3½	18½	58	23	4	21	31	47	29	5/16	1	4½	7	3½	1⅛	7½
7¾	4¾	5½	7⅞	9	7⅝	14¾	2	5	19	39½	20	11	3¾	1¾	2½	107	9	8 11/16	8 13/16	7⅞	8 15/16	17/32	5	32	2½	18½	58	23	4	21	32	47	29	5/16	1	4½	7	3½	1⅛	7¾
8	5	5¾	8⅛	9½	7⅞	15½	2½	5½	21	43	22¼	12	2⅞	1⅞	3	117	10	9	9⅛	8⅛	9¼	9/16	5	33	5	21	64	24	4½	23	33	52	31	⅜	1⅛	5	8	4	1¼	8

FIG. 7. B.E.S.A. STANDARD SHIP'S PROPELLER SHAFT AND STERN TUBE

Extracted by permission of The British Engineering Standards Association from its Report No. 3021, 1922, *Shafting for Marine Purposes*, official copies of which can be obtained from the Offices of the Association, 28 Victoria Street, Westminster, S.W.1. Price 1s. 2d.

4 in. × $\frac{5}{8}$ in. angle. We may order a section made from the 4 in. × 4 in. × $\frac{5}{8}$ in. rolls with the thickness increased to $1\frac{1}{8}$ in. as required. The section produced by rolls designed for a smaller thickness will, of course, be odd but, for a section requiring machined faces, is quite practicable. Fig. 6 shows the section as ordered, the shaded portion representing the section finished machined.

Standard Unit. Where a standard exists for a number of component parts it is more advisable to adhere rigidly to the standard because any slight departure may affect interchangeability and replacement of defective parts.

An example is given in Fig. 7 which shows the B.E.S.A. standard ship's propeller shaft and stern tube which, together with the accompanying table, embodies a sound useful design. It will be evident that general adherence to this standard will, in the event of a breakdown, provide for the rapid ordering and manufacture of quite a number of replace parts of a vital section of the ship's propelling machinery.

The student designer will be advised, when approaching a design new to his experience, to examine the B.E.S.A. list of standards and, if the detail has been standardized, obtain a copy of the publication dealing with the part. Apart from the question of economy involved in using a mature design the reader will often obtain many useful aids to sound practical designs.

EXERCISE

Using Fig. 7 as a guidance, make a working drawing of a stern tube and propeller shaft for a 6 in. diameter shaft. Also, using Fig. 4, Lesson XIII, as a reference, make a working drawing of a bearing block to suit the above shaft.

METALLURGY FOR ENGINEERS

By James Knowles

LESSON XI

THE OPEN HEARTH—(*contd.*)

THE BASIC PROCESS

Of recent years the basic open hearth process has proved an extremely valuable factor in the development of the iron and steel industry. Designed to produce steels of good purity standard from irons of any phosphorus content, it has thus opened up tremendous fields of ore which, so far as most steel specifications are concerned, would otherwise be useless, and furnished an abundant source of cheap, low phosphorus steel scrap for the acid process.

In all open hearth practice the composition and condition of the slag are of vital importance, and the temptation to treat this aspect of the process at some length is strong, but the following notes will perhaps suffice for present needs.

The constituents of a slag should be considered as either *acids* or *bases*, the former being oxides of non-metals, such as silicon, sulphur, and phosphorus, and the latter oxides of metals, e.g. iron, manganese, and calcium. Acids and bases have a strong chemical affinity for each other and the predominance of either type in a slag determines its character and action. Thus, silica (SiO_2) being an acid, a silicious slag will not react with other acids, such as oxides of sulphur (SO_3) and phosphorus (P_2O_5), whereas lime, (CaO) a base, readily enters into combination therewith.

This then distinguishes between acid and basic practice, and indicates the absolute necessity for a furnace lining that is immune from similar chemical action.

During the actual working of the process the condition of the slag needs most careful attention. As in the basic Bessemer, so in the basic open hearth, periodic additions of lime are made with which the phosphorus in the iron, having been oxidized to P_2O_5, combines to form a stable phosphate of lime and, as such, enters the slag. The slag must therefore be kept sufficiently fluid for the maintenance of intimate contact with the metal, basic enough to retain the phosphate, and oxidizing in action so as to fulfil its essential function of impurity elimination. During the final stage, however, it should not be such as to oxidize the *iron*.

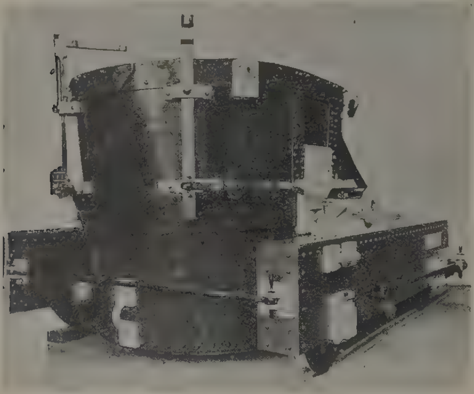


FIG. 18A. DEWHURST'S CARRIAGE LADLE

Since the presence of manganous oxide promotes fluidity, the initial manganese content of a basic charge usually runs about 1 per cent; in fact, a good composition to bear in mind as a typical example is one with silicon, manganese, and phosphorus all at 1·0 per cent.

The Furnace. Owing to the extreme friability of basic bricks, the simple expedient of constructing a furnace entirely of such bricks is automatically ruled out. Furthermore, the obvious alternative of building an outer shell of acid bricks and ramming a basic lining inside would so seriously reduce the capacity of the furnace as to render the arrangement awkward and economically unsuitable. The method of construction adopted is a mean of the foregoing and may be summarized as follows—

1. FURNACE BODY, PORTS AND ROOF. Silica (acid) bricks, mechanically strong and dipped in tar before use.

2. BATH. Stepped magnesite (basic) bricks rammed with burnt dolomite, which is ground and mixed with dehydrated tar and applied in successive layers of about 1½ in. thick, each layer being glazed by heat before another is added.

3. EXTERNAL WORK. Good red brick capable of supporting heavy loads, but used only where low temperatures obtain.

The junction of acid and basic materials occurs slightly above the top level of the bath and, to avoid risk of fusion and collapse along this line at high furnace temperatures, it was customary to insert a neutral insulating band of chrome iron ore between. This expensive precaution has now been abandoned owing to the improvement in the quality of refractories, and recognition of the fact that fluxing does not take place if pressure on the bricks is minimized. Consequently, furnaces are designed so that the weight of the roof is not borne by the side walls, but is supported by a long horizontal arch, or carried on special girders.

In all open hearth furnaces large amounts of dust from the ore and other materials charged, together with splashes of slag, are carried away with and by the waste gases, and in basic practice, where the relative quantity of slag produced is considerable, this reaches serious proportions. Therefore, to prevent choking of the regenerators and fluxing or glazing of the chequer bricks, supplementary chambers are built between the ports and regenerators in which this slag and dust are deposited and collected. *Slag pockets*, or *dust catchers* are an essential feature in all basic furnace design, but also have been widely adopted for the larger acid furnaces.

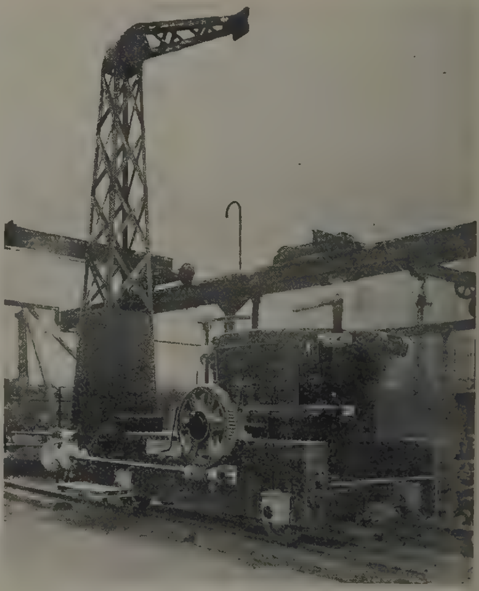


FIG. 18B. ELECTRICALLY-PROPELLED CARRIAGE LADLE

Working the Charge. The furnace is soaked with heat and the bottom liberally covered with ore and lime for the rapid formation of a basic slag. The pig-iron is charged gradually, small lime and ore additions being often made concurrently, followed by the scrap. Melting then proceeds as quickly as possible, feeds of ore, or other oxides, and lime are given after the metal is clear melted and the boil gently urged forward.

Here an important point must be stressed. In the working of the basic process it has been found practically impossible to eliminate the phosphorus before the carbon is almost removed. On the other hand, with the carbon eliminated, the bath goes off the boil and no further reduction of phosphorus percentage can be effected. This latter contingency, obviously most likely to arise when the phosphorus content of the charge is high, is met by the immediate introduction of a small quantity of low phosphorus pig into the bath. By these means a very small percentage of carbon is added and the boil recommenced. It will be realized then, that though the usual, regular series of bath samples are taken, inspected and rapidly analysed for both carbon and phosphorus, reactions cannot be stopped at any desired carbon point with the certainty of a low phosphorus content in the finished steel. Furthermore, if the bath were run right down and some recarburizing agent added, the reducing action of the carbon would effect a dissociation of the lime phosphate in the slag and a reprecipitation of phosphorus into the bath. Therefore, the basic process is employed primarily for the production of mild or dead soft qualities, although higher percentages of carbon can be obtained by recarburizing *in the ladle* during tapping with, for example, anthracite. For the same reason, deoxidizing agents are also fed into the stream of molten steel or ladle.

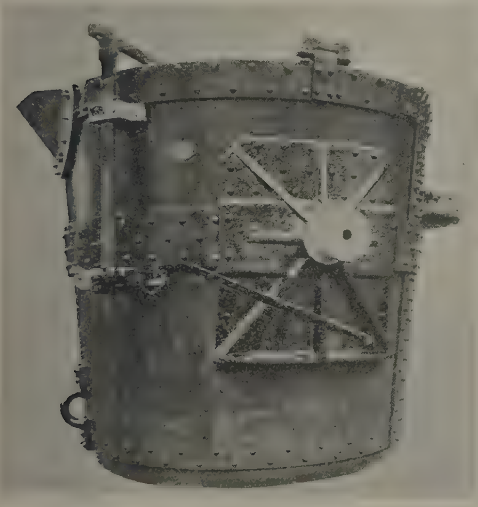


FIG. 18C. LADLE FOR OVERHEAD CASTING

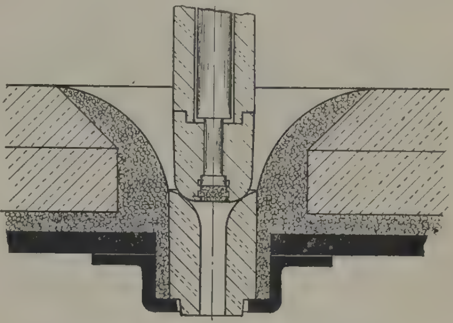


FIG. 19. SECTION THROUGH STOPPER AND NOZZLE

(Steel casting ladle)
The Park Gate Iron & Steel Co., Ltd.

The Talbot Process.[1] The continuous process invented by Mr. Benjamin Talbot, though of comparatively recent origin (1900 *c.*), is probably the most important commercial application of the basic open hearth principle, and has met with remarkable success in the production of steel for rails and structural purposes. Almost universally the employment of a tilting furnace of very large capacity (150–250 tons) is involved, and the charge introduced in a molten condition.

Tilting furnaces are built on rockers or rollers and, being entirely separate from the blocks, tilt or rotate about an axis passing roughly through the lower ports. With the furnace in the vertical position the tap hole is situated slightly above the bath, and at the charging side slag spouts are provided at a suitable level. Hence, by tilting the furnace in the requisite direction, either the finished charge or slag can be drawn off as, when and *in any quantity* desired.

The first heat is worked in the ordinary way until, in condition for tapping, when *a predetermined proportion* (usually about one-third) is run into the ladle, recarburized if necessary and deoxidized. The furnace is then tilted backwards for slag removal and finally restored to the vertical. A fresh slag is made by the addition of lime and oxides and a further charge of "hot metal" introduced.

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[1] See Plate facing page 1209.

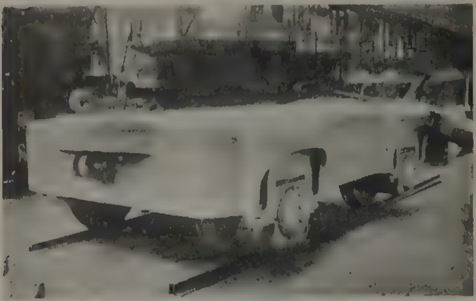


Fig. 20. Dewhurst Ingot Bogie for Car Casting System

Designed to carry 3 10-ton or 6 3-ton ingots

The salient features of the process should now be apparent. Firstly, a considerable amount of purified metal and reserve of heat always remain in the furnace and, therefore, the charge introduced suffers considerable dilution in regard to its removable elements. Secondly, since a basic and active slag is already present when fresh supplies of metal are charged instead of being gradually formed afterwards, intimate contact therewith is immediately effected. The metaloids are at once attacked, the "melt" is worked down through all the familiar stages, a good finishing slag formed, and again a portion of the metal tapped off.

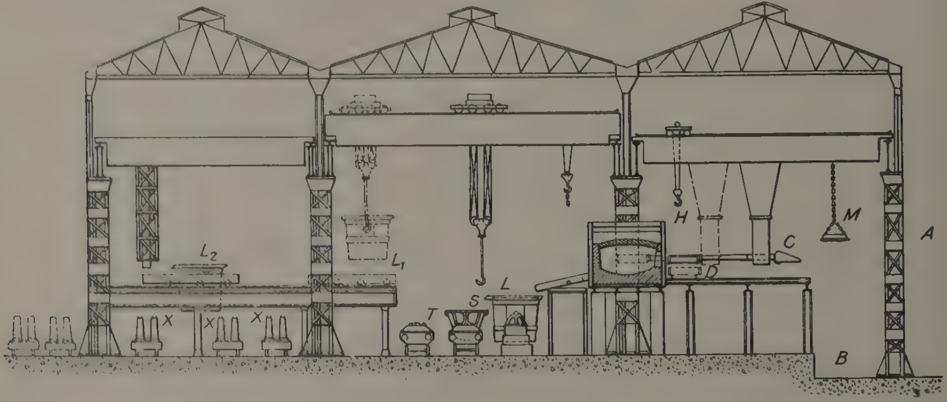


Fig. 21. Layout of Modern Open Hearth Plant by Dewhurst's Engineering Co., Ltd., Sheffield

Note continuous forward movement. Charging boxes are brought from assembling bay *A* to position *B*, and hoisted on to working stage at *D* ready for charging by electric overhead charger *C*. This is constructed on a double gantry which also accommodates a magnet crane *M* and auxiliary hoist *H*. Tapped steel enters ladle *L*, slag overflowing into slag ladle *S* and refuse from under furnace collected in a transfer car, similar to *T*. Casting ladle is subsequently conveyed to position *L1* by casting bay crane, drawn to *L2* on electric ladle car, and ingot moulds *X* brought underneath, on cars, for teeming

So the process continues until the week-end, when the furnace is emptied, inspected, fettled, and all preparations made for the commencement of another run.

Practical results coincide very closely with the indicated theoretical expectations of rapid working, huge outputs, and low fuel costs. The metal charged, however, should be fairly low in silicon, so that the boil may be obtained quickly, and if a regular supply of suitable iron is uncertain of obtainment from the blast furnaces, some desiliconizing must be done in the mixer. This should, therefore, be capable of fulfilling the rôle of preliminary furnace and of sufficient capacity as adequately to supply the needs of all furnace units depending on it. For convenience it should be situated in the middle of the furnace line.

WORK IN THE CASTING BAY

Ingot Moulds for both Bessemer and open hearth processes are composed of low phosphorus hematite iron, so as to minimize the risk of premature cracking, and also to bring them within the first-class scrap category after their period of usefulness has expired. Internally, they may present rectangular, octagonal, circular or some composite form of cross-section, and vary in dimensions according to the weight of ingot desired. Considering the common rectangular form, for weights of 20 to 60 cwt., the heights may run from 5 ft. to 5 ft. 6 in., and maximum widths from 12 in. to 24 in. To facilitate stripping the sides are tapered, usually by about 2 in., with the wider end at the top or bottom as specified. In the latter case the moulds are fitted with staples, or lugs at the top, and are lifted off the solid ingot, whereas in the former, termed *inverted* moulds, a mild steel hook, or staple is set in the solidifying ingot by which means it is afterwards drawn out of the mould. Moulds are generally open at both ends and, when the casting pit is being "set up," are accurately placed on bottom plates and luted up with gannister.

The Casting Ladle is lined with a double course of fire-bricks, and may either be mounted on a bogie carriage or slung from an overhead crane. The molten steel is run out from the bottom of the ladle which is designed to drain towards the *nozzle,* which is fire-clay lined and cup-shaped at the top. The *stopper* is an iron rod round which, before each cast, fire-clay sleeve sections, dovetailing into each other, are carefully screwed and fitted together, the joints being made good with gannister or fire-clay. The base of the bottom section, or *stopper end,* is semi-spherical in shape, and secured to the stopper rod by a countersunk bolt and cotter pin. This accurately engages in the cup of the nozzle and, after careful adjustment, forms an effective valve. The top of the stopper is connected by a *goose neck* to a hand-operated lever by which it is raised or lowered, and the stream of steel liberated or shut off. Generally a second nozzle and stopper are provided, either to act as a stand-by in case of emergency, or to permit of the simultaneous teeming of two ingots.

By the courtesy of Messrs. Dewhurst's Engineering Company, Ltd., Sheffield, photographs of three types of casting ladles are herewith reproduced. Fig. 18A represents the ordinary type of carriage ladle which is towed over the ingot moulds by locomotive crane; Fig. 18B an excellent example of the employment of electric propulsion, and Fig. 18C a ladle used for overhead casting. In all these, although the rods are not in position, duplicated stopper gear may be noted.

Casting complete, before stripping, the ingots are left to solidify for a period dependent on their size and quality; 20 to 30 cwt. mild steel ingots may safely be stripped an hour after casting, but larger sizes, or stiffer qualities, should stand at least twice as long.

Should an ingot stick so that it cannot be dislodged from the mould by brute force, bad temper, and a sledge hammer, it is taken to a *stripper.* Strippers are of many types, either in the form of hydraulic, ground fixtures or electric overhead crane auxiliaries, but all work on a similar plan—the ingot is held fast whilst the mould is pressed off, or vice versa.

In many mills ingots are charged into reheating furnaces whilst still hot and, whenever possible, these furnaces are built so closely adjacent, or parallel to the casting pit, that they are within the span of the stripping crane. There are cases, however, where owing to local conditions, they are situated some distance from the pit side. Now, curtailment of the time taken in stripping and conveyance is essential on account of the importance of conservation of heat. To this end the *car casting* system is adopted in many large plants. Moulds are set up on bogies and, the casting ladle being slung from an overhead crane, teeming need not, of necessity, be conducted beside the furnace. This considerably eases congestion, but is not universally adopted for, sometimes, the cars are run into a sunken road and filled from the ordinary ground type ladle. In either case, however, directly the ingots are stripped they are run direct on the bogies to the mill, but it is obvious that if inverted moulds are employed a second train must be supplied for the ingots.

(CONCLUSION)

BLAST FURNACE PRACTICE

By Percy C. R. Kingscott, D.I.C., A.R.C.Sc., F.I.C., B.Sc. (Lond.)

LESSON XIII

CALCULATION OF THE BURDEN

Slags cannot be calculated from chemical laws; they are largely empirical. By long experience, furnace operators have discovered that certain ratios of lime to silica with varying proportions of alumina and magnesia have certain effects under known conditions of furnace practice. For example, a ratio of bases to acid of 1·3 : 1 is usually suitable for foundry iron, 1·5 : 1 for basic iron, other grades lying between these ratios.

If the phosphorus content be unimportant, the calculation of an ordinary slag is fairly straightforward. We have already seen that for a given basicity the amount of limestone required by an ore is easily determined. It must also be presumed that we know the efficiency of the limestone or mixture of limestone and dolomite, as well as the available carbon and analysis of the ash of the coke or mixtures of cokes. In other words, it is absolutely essential to have every material properly sampled and analysed. In the olden days much trouble was caused by the rule of thumb methods of the operators who merely shrewdly guessed at the probable nature of the ores, etc. Nowadays, rigid analysis is an axiom. In burdening, it is necessary moreover to base the figures upon analysis "as received," and not of the dry materials.

It is customary to charge always the same weight of coke in each operation, this being limited by the size of the furnace hopper, charging apparatus, or personal views of the operator. The weights of ores and limestone are varied according to requirements. Having the weight of the coke charge and the coke consumption or ratio of ores to coke, the latter figures being generally obtained by experience, it is then necessary to calculate a mixture of ores and limestone which will flux to produce, with the coke ash, a slag of prearranged basicity. The operator must remember that the sulphur present should pass away chiefly in the slag, the maximum quantity carried being seldom more than 2 per cent. He must, therefore, arrange his mixture so that the quantity of slag produced is sufficient to absorb the sulphur liberated, at the same time selecting his materials in proportions which will produce a required maximum percentage of phosphorus in the pig. Slags, as a general rule, should not be less in quantity than 50 per cent of the weight of the pig-iron produced.

Burdening by Furnace Inspection

To an experienced smelter, the appearance of the slag conveys much valuable information concerning its composition, the iron expected, and the temperature of the hearth. It also warns him of furnace irregularities which must be diagnosed and checked. Slags containing a large percentage of bases have a greyish blue granular appearance when cold. A high content of lime generally causes the slag to slake, on standing for a while. A diminution in the quantity of bases causes the slag to take on a vitreous appearance which first shows itself as a glassy edge. As the acid constituents increase, the slag becomes glassy throughout, and the colour then shows whether the furnace be hot or cold. Cold acid slags are generally dark or even black, and show that the furnace is deranged by overburdening, shortage of lime, leakage of water, or by charging poor coke. A black calcareous slag also denotes a cold hearth, probably due to poor coke, furnace slipping, leakage of water or other causes.

Siliceous slags tend to draw out into fine strings before solidifying; limy slags are short. The former slags tend to froth, whereas the latter sink quietly into the slag ladle.

The operator, by noticing the conditions of the slag, the appearance of the gas and fume, the running iron, the condition of the burning coke when viewed through the tuyere peepholes, the blast pressure and other factors, and by then exercising his judgment, increases or decreases the burden, increases or decreases the limestone or basicity of the slag, alters the blast temperature, or alters the volume of the blast according as the trouble is temporary or lasting.

It is obvious that the quickest remedy is the alteration of the temperature of the blast, whereas changes in the burden or basicity must take time to show their effect in the hearth. When working with standard materials, burdening as described is sufficient, provided the materials used are constantly analysed and any serious variations immediately taken into consideration.

When dealing with new problems in burdening introduced by strange ores, or those in unfamiliar proportions for the manufacture of fresh grades of iron, or in blowing in or damping down a furnace, it is essential to calculate the burden and then to correct the results by observation.

THEORETICAL BURDENING

It is first necessary to decide upon the proportion of ingredients of the mixture of ores which are available. This will depend upon several factors, such as availability and physical and chemical properties. Having decided upon the mixture, the average analysis can then be calculated, and also the amount of flux necessary to deal with this average analysis. This calculation is done in precisely the same manner as that previously given for ore valuation, whereby the amounts of flux and coke are obtained. The basicity of the slag is assumed according to the grade of iron, amount of alumina present, and general conditions of working pertaining to the particular practice or furnace.

There are other methods of burdening, such as predetermining the ratio of ores to coke necessary to suit certain conditions. In this case, the ore mixture is made up in exactly the same manner, and the flux calculated as before. Care must be taken to allow the requisite amount of limestone for the coke ash, this being usually constant for a particular charge of coke of known analysis and for a given basicity of slag.

It is general to allow a certain loss of iron in the slag and flue dust, but in good practice these losses are kept at a minimum, and owing to the carbon, manganese, silicon, sulphur, and phosphorus in the pig, the weight of the latter recovered should be at least 100 per cent of the weight of iron introduced in the ores and coke ash.

The analyses of the slag from these mixtures are then set out and the mixture altered according as the alumina, sulphur, or other material content is wrong.

Sometimes the ores are more than self-fluxing, i.e. where the material to be fluxed is lime and not silica. For example, we may have ores and coke available with the following percentage analyses—

	A	B	Coke
Fe . .	30	45	1
CaO .	18	—	1
Al_2O_3 .	2	5	2
SiO_2 .	10	16	7

Let it be assumed that the coke charge is 5 tons, and that the coke consumption is 25 cwt. per ton of pig. The coke charge will give 4 tons of pig-iron, which, assuming 100 per cent recovery of pig-iron, is equivalent to 4 tons of iron required in the ore charge.

Using 3 parts of A and 1 part of B in the ore mixture will give an average iron content of 33·75 per cent, and for the 5 ton coke charge, will take 8·89 tons of A and 2·96 tons of B.

Assuming 2 per cent Si in the pig accounts for 0·171 tons of SiO_2 from the charge.

	Tons
The total silica from ores and coke ash is	1·713
Silica reduced to silicon	·171
Silica to be fluxed	1·542

With a ratio of lime to silica of 1·4 : 1, this silica requires 2·159 tons of lime. The lime available from the ore A and coke ash is 1·65 tons, leaving 0·51 tons to be added in the form of limestone. At 51 per cent efficiency, the amount of limestone required is 1 ton per charge.

	SiO_2	Al_2O_3	CaO
Ore A . .	·889	·178	1·60
Ore B . .	·474	·148	—
Coke . .	·350	·100	·05
Limestone .	—	—	·51
Less	·171	—	—
	1·542	·426	2·16

The total of these compounds in the slag is 4·128 tons. Assuming this weight comprises 93 per cent of the slag, allowing 7 per cent for magnesia, alkalis, etc., the percentage amounts of the above ingredients of the slag are as follows—

	Per Cent	Ratio
SiO_2 . .	34·7	
Al_2O_3 . . .	9·6	$\frac{CaO}{SiO_2} = 1·40$
CaO . . .	48·7	
Etc. . . .	7·0	
	100·0	

Naturally, by lowering the proportion of ore B in the mixture, the use of limestone can be avoided, the mixture then becoming self-fluxing. This practice obtains in North Lincolnshire and in the Birmingham district, Alabama.

The ratio of ore to fuel depends upon judgment and is adjusted to what the furnace can be made to carry. Much depends upon the volume of slag to be melted, but the furnace performance, after conditions have been set, supplies the best information.

DEVELOPMENT OF MODERN LINES

The blast furnace recommended by Bell, in 1840, had the following dimensions—

	Ft.	In.		Ft.	In.
Furnace height .	80	0	Height of hearth	6	0
Diam. of hearth .	8	0	Height of bosh .	18	0
Diam. of bosh .	20	0	Diam. of throat	11	0
Angle of bosh .	72°				

and, as shown by Brassert, this shape, with a somewhat wider throat, was typical of American furnaces 40 to 50 years after. Since that date, boshes have been lowered, the hearth diameter has been increased, but the furnace height and diameter of throat have not been altered to so great an extent. The angle of the bosh has in the meantime changed from 79° down to about 72° again and then, later, up to about 80°. The dimensions of No. 4 furnace, South Works, Illinois Steel Company, as given by Brassert (*A.I.S.I.*, 1914), are shown in the table.

According to McKee (*Proc. Eng. Soc., W. Penna*, 1926, 41, 396), the furnace section has since deviated somewhat from its previous cylindrical tendency, in that the hearth and bosh diameters have been further increased. The lines of No. 5 furnace, 1924, Illinois Steel Co., are taken from this paper as an example. (See Table.)

Bell's furnace dealt with rough ore, and was driven at a very slow rate with comparatively cool blast. With the lowering of the bosh, output increased up to a certain rate, but at this point, approximately in the year 1908, serious difficulties in operations arose owing to the flat bosh. The increase in hearth diameters which followed, coincident with increased blowing power, rendered much larger outputs possible, the steeper angle of the bosh removing many of the difficulties of operation. McKee attributes the smoother achievement of large outputs not only to these factors, but also to the improvements in blowing machinery and stove efficiencies and temperature. No. 5 furnace is typical of furnaces which, using dusty ores, have a fairly large flue dust loss, through producing large outputs regardless of the expense of such loss. These furnaces are in some cases blown at 60,000 cub. ft. per minute. McKee states that the lines of the furnace of the Shenago Co. are typical of those furnaces which, although working fine ores, have a minimum flue dust loss, owing to blowing a less volume of blast which, however, is heated to a higher temperature than is the case of the furnace typified by No. 5 furnace. The Shenago furnace is blown at the rate of 38,000 cub. ft. per minute at a straight line temperature of 1,500° F., and produces 600 tons of iron per day with a coke consumption of about 1,850 lb. per ton of iron.

Mussey describes No. 1 furnace, Woodward Co., Ala. (*Trans. A.M.I.M.*, 1924), and Keenan, "C" furnace, Tata Co., India (*J.I.S.I.* cxii, 1925), both furnaces giving large outputs with economical working in their respective practice.

	No. 4 Illinois		No. 5 Illinois		Shenago		Woodward		Tata	
	ft.	in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.
Height of furnace	90	0	92	0	94	0	90	8	89	2
Hearth diameter	17	8	21	6	19	6	17	6	17	0
Bosh diameter	22	0	24	3	23	3	21	6	21	6
Throat diameter	17	0	17	0	16	6	16	0	16	0
Bell diameter	—		13	0	12	6	12	0	12	0
Hearth height	9	9	9	9	9	0	10	6	8	4
Bosh height	12	3	8	3	10	$7\frac{1}{2}$	12	$3\frac{1}{2}$	11	8
Bosh cylinder height	7	0	7	3	17	$4\frac{1}{2}$	11	0	8	4
Angle of bosh	80°		80°32′		80°1′		79°45′		79°6′	
Batter of stack	$\frac{3}{4}$ in. per ft.		1 in. per ft.		$1\frac{1}{8}$ in. per ft.		$\frac{55}{64}$ in. per ft.		$\frac{25}{32}$ in. per ft.	
Top cylinder depth	21	0	23	3	21	1	18	$7\frac{1}{2}$	18	8
Height of stack inwall	40	0	43	6	35	11	—		—	

Modern British Furnaces. The trend of British practice as exemplified by four furnace plants typical of different home practice, is shown in a schedule prepared by Clements (*Emp. M.M. Cong.*, 1924). The lines of these furnaces are given in the table below.

It is felt among certain operators that a low, steep bosh and wider hearth are likely to give smoother working of the furnace. There is a tendency also to obtain large outputs by the use of prepared materials and with larger blast volumes capable, if necessary, of being delivered at higher pressures than those, which, in older practice, were considered the maximum for safe and economical working.

	Carnforth Co. No. 1 Furnace		Dorman Long "C" Furnace		Ebbw Vale No. 5 Furnace		Parkgate No. 3 Furnace	
	ft.	in.	ft.	in.	ft.	in.	ft.	in.
Height of furnace	80	0	80	6	88	10	71	0
Hearth diameter	12	6	13	0	14	0	11	0
Bosh diameter	20	2¾	21	6	22	0	20	0
Throat diameter	15	6	16	9	15	6	12	9
Bell diameter	9	6	11	0	11	0	9	0
Hearth height	9	0	10	0	9	4	8	6
Bosh height	13	6	14	6	13	0	13	6
Bosh cylinder height	—		—		6	0	—	
Angle of bosh	75°		73°		73°		68°	
Top cylinder depth	10	0	10	0	13	6	—	
Height of stack inwall	47	6	31	2	45	0	49	0

(CONCLUSION)

MACHINE TOOLS

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XXI

THE SCREW MACHINES

These are provided with a turret, and have many other features in common with the turret lathes. The screwing machines are a wholly different class, used only for threading. The specific term "automatic" without qualification denotes a particular group of screw machine designs only. But all screw machines are not automatic. The automatic turning machines are a more recent group, the functions of which are very restricted, in regard both to the range of operations and dimensions, but which are employed in a rapidly increasing extent for quantity production.

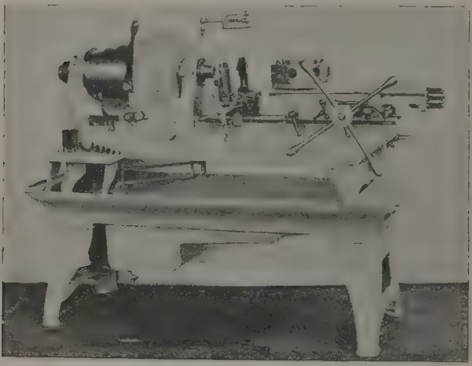


Fig. 149. Plain Screw Machine

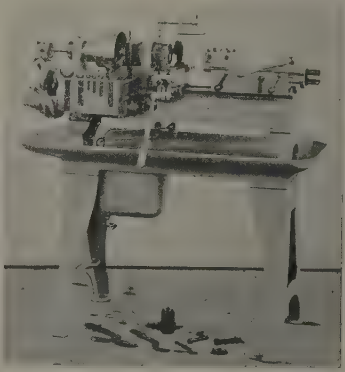


Fig. 150. Wire Feed Screw Machine

The Plain Machines. Screw machines are divisible into three groups—plain, wire feed, and fully automatic. Each type is illustrated with photographs by the Brown & Sharpe Manufacturing Company.

Fig. 149 is a plain machine, in which all work must be chucked by hand, either in the front, or from bar passed through the hollow spindle. It is a design of much utility within its own sphere, which is that of small pieces in moderate quantities. Driven with a 3-stepped belt cone, it has back gears operated with friction clutches, which, with the simple belt drive, give twelve changes of spindle speeds direct from 35 to 545 r.p.m. in geometrical progression, and six reverse speeds from the countershaft from 76 to 545 r.p.m. The frictions are put in with a lever in front of the headstock, through a sliding collar and cams, which, when making engagement with one, releases the other instantly.

The round turret has eight holes, each with its independent tool stop in a disc seen at the rear of Fig. 149. The feed of the turret is automatic, being communicated through a chain that comes direct from the spindle. For each spindle speed, eight changes of feed are obtained with cones of change gears, from ·003 in. to ·034 in. in geometrical progression, per revolution of spindle. These are controlled with the small lever seen below the headstock. The turret is fed with a rack down the centre of the slide bed, and is automatically clamped each time it is moved forward. The thrust of heavy cuts is resisted by a stop dog in the turret slide bed engaging with a rack on the bed of the machine. The cross slide can be clamped anywhere on the machine bed. It has screw stops for both front and back tools, set with dials to half-thousandths of an inch.

Wire Feed Screw Machines. A Brown & Sharpe screw machine of this type is shown by Fig. 150, and a section through its spindle by Fig. 151. The stock bar is held in the spring collet at the front—entirely enclosed, and which has no longitudinal movement. It is closed with the encircling tapered sleeve, operated by the levers at the rear, which are thrust along by the collar *A*, slid with a lever that extends below the bed. The bar stock is advanced through the spindle by means of the spring feeding finger located directly behind the spring collet, and screwed to a tube that extends through the spindle to the rear, the wire feed.

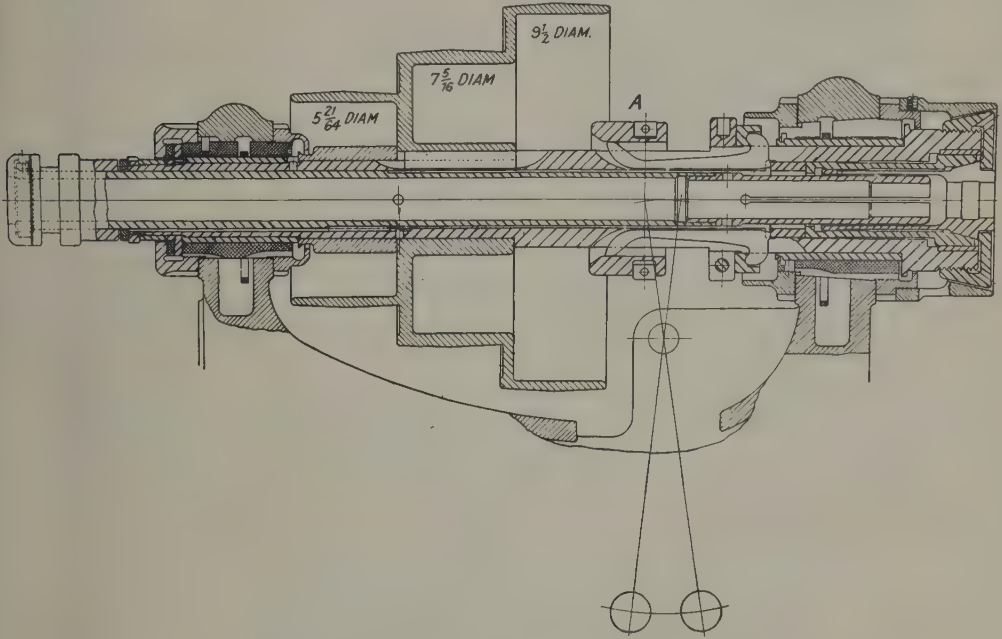


FIG. 151. HEADSTOCK OF MACHINE IN PREVIOUS FIGURE

The chuck is opened, the stock bar advanced, to the length required, and the chuck closed on the bar by the automatic action of the machine, the chuck and wire feed being operated in conjunction. But if a latch is lifted in the feed tube the feed will be thrown out. The cam seen at the left of the headstock in Fig. 150 actuates the feed, another cam, not seen in front opens and closes the chuck. The length of feed can be changed with a sliding block seen behind the feed cam.

This machine is made in five different capacities, with variations in the fitting of automatic chucks and cross slide details. The one shown in Fig. 150 has six changes of spindle speed, direct, from 150 to 1,400 r.p.m. in geometrical progression, and three reverse from 573 to 1,400 r.p.m. These machines are suitable for dealing with small and moderate quantities. They require no elaborate set-up, but must have constant attention. For very large outputs the fully automatic screw machines are preferable. They are much more complicated, but when once they set up they need no attention. They will perform all the operations done in the turret lathes—the tools being in the main identical.

The Automatic Screw Machines. One hardly knows how to deal with this group, which would require a treatise for a full description. The photographs and drawings must largely explain themselves. Figs. 152 and 153 are front and rear views respectively of one of the Brown & Sharpe machines, and Fig. 154 is a section through the spindle, with figure references.

It will be noticed that the turret is mounted with its axis horizontally. The object of this is to allow the tool in position to work closely in between the cross slide tools with the minimum of overhang, while the idle tools are in such a position that they do not interfere with the cross slides. This also permits of the use of a long bearing, since the taper shank of the turret extends through to the rear of the slide. The turret is revolved with a roll engaging radial grooves in a disc attached to the turret shank. The effect is to produce an accelerating and retarding action, which results in rapid indexing without shock. It is automatically locked in position with a hardened taper pin in a hardened bushing.

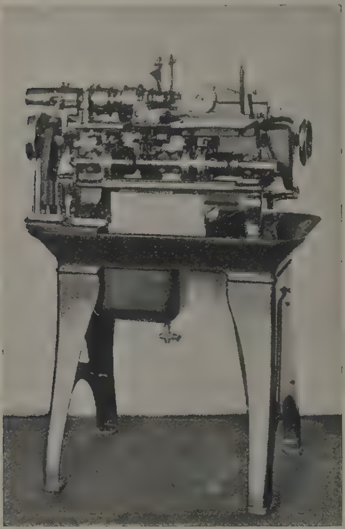


FIG. 152. AUTOMATIC SCREW MACHINE

All feeds and movements of the machine other than the revolution of the spindle are driven from the feed shaft at the rear of the bed, Fig. 153. This shaft is belted from the overhead works, and runs at a constant speed, irrespective of that of the spindle. The movement of a lever at the front near the spindle stops the feed without arresting that of the spindle. All operations are timed or regulated from the camshafts on the front end of the bed. These are driven from the constant speed shaft at the rear through a worm, a worm wheel, and change gears, at the end of the bed. The change gears give the duration of the cycle of operations or the length of time taken to produce one piece. The spindle is driven from overhead with changes produced with belted cone pulleys, for which a single pulley running at a constant speed with variations obtained by means of change gears are substituted. Twelve changes are obtained in geometrical progression. The operations of feeding the stock, reversing the spindle, and indexing the turret are regulated and varied with trip dogs mounted on carriers on the camshaft. The dogs actuate levers that extend through the bed of the machine to the rear shaft, where they connect to positive clutches. The tripping of a lever engages the corresponding clutch, causing gears, or a path cam to revolve with the feed shaft. After the performance of a single operation the clutch is automatically released. The turret slide and cross slides are advanced to the work by the action of disc cams and levers. One, two, or three pieces of work may be completed in one revolution of the cams. The series of curves on the edge to make one piece will occupy the whole circumference, or will be repeated according to the number of pieces to

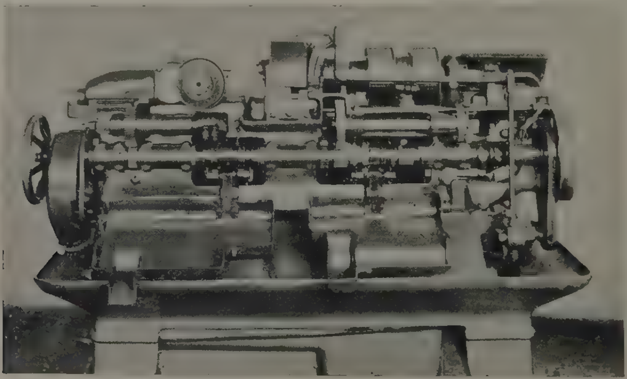


FIG. 153. REAR OF AUTOMATIC SCREW MACHINE

be produced in a revolution. The idle movements of withdrawing and advancing the tools, feeding the stock, and indexing the turret are independent operations schemed to reduce to a minimum the non-cutting time.

Movements are timed positively by the direct action of cams, which have no lost motion. The edges have to be laid out on steel discs, and machined to any regular or irregular outlines. The preparation and the set-up are tedious, so that a large product is necessary. For each separate job, cams have to be redesigned and dogs re-arranged. But once set up, the sequence of operations is entirely automatic. When work is duplicated at intervals, the same sets of cams can be taken from stock and used again, with the certainty that parts from each batch will interchange. The turret and cross slide tools can be made to operate in conjunction owing to the accuracy with which the cams are laid out, and the fact that all three cams are rotated at a constant speed. This co-ordination permits of the use of various tools fed to the work by the combined action of the turret and cross slides. Examples of this kind are swing tools for turning, knurling, thread rolling, behind shoulders or recessing within a hole, tools for turning tapers or cutting off at an angle.

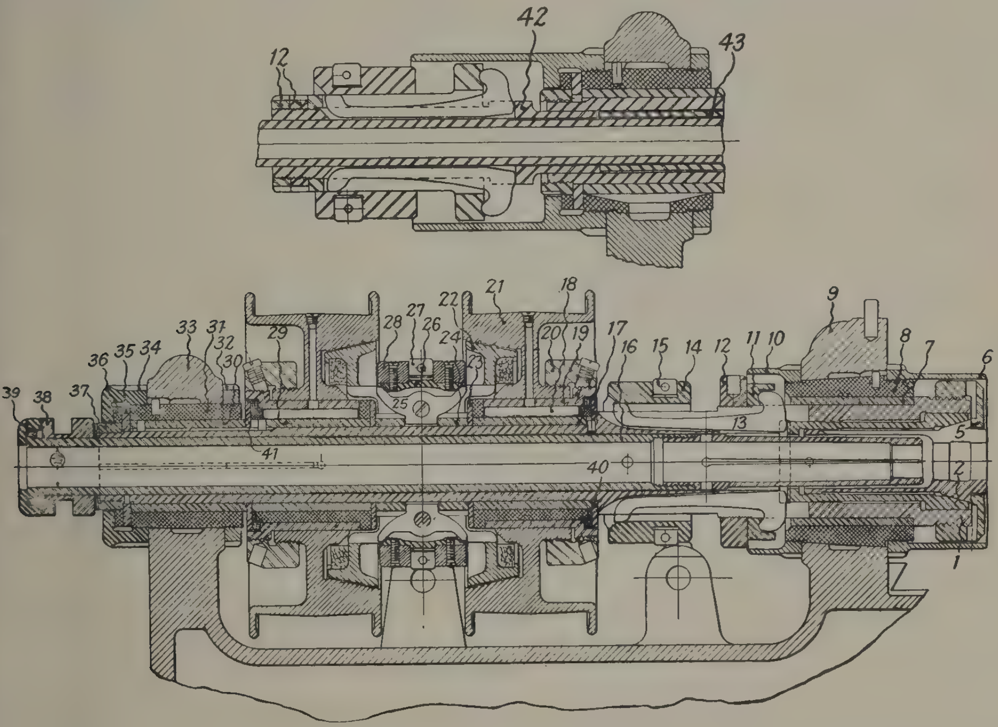


FIG. 154. THE SPINDLE MECHANISM

1. Spindle.
2. Chuck sleeve.
5. Chuck nut.
6. Spindle front box nut—right.
7. Spindle front box sleeve.
8. Spindle front box.
9. Spindle front box cap.
10. Spindle front box nut—left.
11. Chuck lever fulcrum.
12. Chuck adj. nut.
13. Chuck lever.
14. Chuck fork sleeve.
15. Chuck fork ring.
16. Feed tube.
17. Spindle pulley sleeve—right.
18. Roller bearing.
19. Spindle pulley bushing.
20. Spindle pulley adj. nut.
21. Spindle pulley.
22. Clutch body.
23. Spindle pulley thrust washer.
24. Clutch body bushing.
25. Clutch lever.
26. Clutch shoe.
27. Clutch sleeve ring.
28. Clutch sleeve.
29. Spindle pulley sleeve—left.
30. Spindle rear box sleeve.
31. Spindle rear box.
32. Spindle rear box nut.
33. Spindle rear box cap.
34. Spindle thrust nut.
35. Spindle thrust washer.
36. Spindle thrust washer.
37. Spindle nut.
38. Feed tube collar.
39. Feed tube nut.
40. Spindle pulley thrust wash. Spindle pulley nut—right.
41. Spindle pulley thrust wash. Spindle pulley nut—left.
42. Chuck lever fulcrum shoe.
43. Chuck closing tube.

The Spindle Mechanism. Fig. 154 shows this in longitudinal section. The stock is advanced by means of the feed tube, No. 16, operated with cam mechanism—not shown, at the rear end. The cam is of the path type like that in Fig. 150. By means of a sliding block, the position of which can be varied, the length of feed is regulated by the aid of a scale, graduated to thirty-seconds of an inch, or to millimetres. The spring collet, which has no endlong movement, is closed by the internal taper on the sleeve 2, its grip being regulated with the adjusting nut 12. To change collets of different sizes the nut 5 is unscrewed and removed. The sleeve is tightened on the collets by means of the levers 13, actuated by the sleeve 14 with a ring and cam. The advantage of locating these levers close behind the collet is, that it permits of using up stock bars close to the end. In the smaller machines, the levers are located at the rear, as shown in the small detail in Fig. 154.

The operation of the two pulleys, one for driving, the other for reverse, is effected with conical friction clutches. Levers, 25, prevented from longitudinal movement, and rocking about a central pin have their horns thrown over in one direction or the other by the lateral sliding of the steel-faced clutch sleeve 28, which thrusts one or other of the clutch cones 22 into engagement with its spindle pulley. The pulleys run on roller bearings.

Machines with Constant Speed Drive. To abandon the belted cones on countershafts for changing spindle speeds, and substitute a single pulley running at a constant speed on the machine, or a motor, is an alternative in these machines which has been noticed in connection with other types. The pulley makes 400 r.p.m. A friction clutch within the pulley, and operated with a lever at the front serves to start and stop the machine. A brake is connected with the lever, so that one movement throws off the power, applies the brake that stops the spindle and locks the driving mechanism.

Subject to slight variations in different machines, the following is the method of transmitting power from the pulley shaft to the spindle. In Fig. 155—a cabinet design, the pulley shaft and spindle mechanism are enclosed in the base. Variations of spindle speeds are obtained with change gears. When the thimble *A* engages the constant speed pulley *B*, power is transmitted through the pulley shaft and change gears *E* to the sprockets *D* and *G*. The sprockets *F* and *H* are mounted on sleeves, and run loosely on the driving shaft, and are driven with chains from *D* and *G*. The friction clutch *K*, by engaging *F* or *H*, transmits power to the spindle-driving pulleys *I* and *J* which are pinned to the drive shaft. Two forward and two backward speeds, one fast and one slow for each combination of back gears can be obtained with a friction clutch controlled by trip dogs. Twelve to fifteen changes of spindle speed are obtained in different machines. The general construction of the machines other than that of the spindle is as previously described.

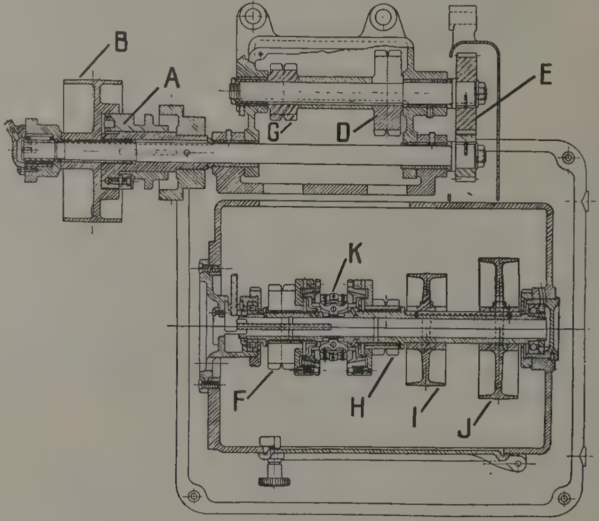


Fig. 155. Spindle Driving Arrangement

PATTERN-MAKING AND FOUNDRY WORK

By J. McLachlan and C. A. Otto

LESSON XIII

MACHINE MOULDING

Considerable improvements have been made during recent years in the production of mechanical appliances for making moulds, and they have been instrumental in speeding up production and reducing manual labour.

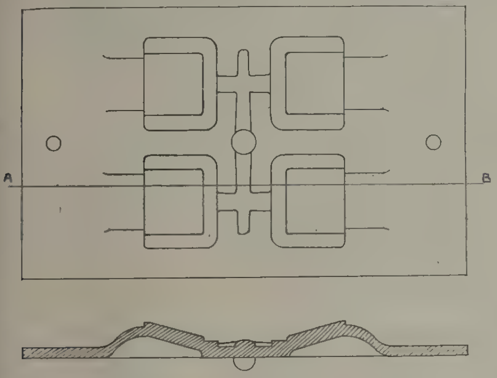


Fig. 64. All-cast Turnover or Match Plate

Primarily, moulding machines are designed to cope with work of a repetition character, and, with few exceptions, they are not likely to be economically employed unless the number of castings of similar design warrants the additional outlay in preparing a first-class pattern for use with a machine. The exceptions refer particularly to those machines that only ram the sand in moulds, and which are usually referred to as moulding machines. Generally, it may be assumed that moulding machines are more applicable for repetition work, and the greater the number of castings required for any one design, the lower will be the cost per casting produced, because the high cost of pattern work, suitable for use on a machine, will be spread over a large number of castings and becomes less appreciable.

In addition to greater speed of production from moulding machines, the resulting castings are more accurate, cleaner, and possess a more regular surface than is obtained from hand-made moulds. The reduction in the amount of rapping necessary, because of the accuracy of stripping appliances, reduces any variance in weight, between castings from the same pattern, to a minimum, and, as no patching is necessary, the castings are well defined and an exact counterpart of the patterns used.

Types of Machines. Moulding machines may be classified as hand and power machines, and each class may be divided into various types. The operations performed by the machine are somewhat similar in each class. The hand machines, for instance, may be either hand, press, or jolt-ramming machines. Invariably, each of these types has facilities for mechanical stripping by some form of lever appliance capable of being operated by hand, and, in addition, some can be fitted with turnover attachments and draw frames.

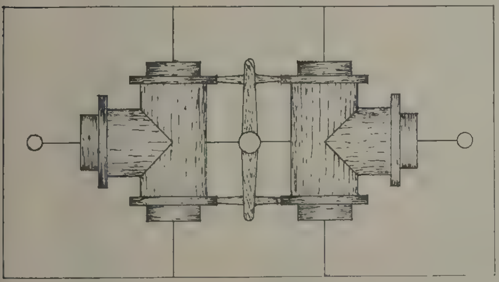


Fig. 65. Reversible Pattern Plates

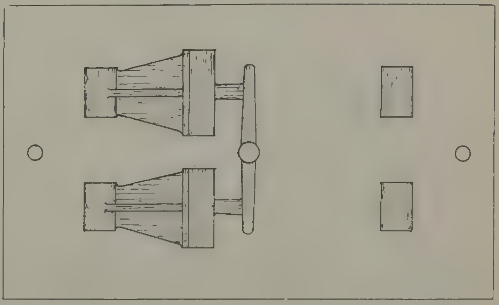


Fig. 66. All-cast Reversible Pattern Plate

The hand machines have a special value in that they do not involve the expense of installing power plant and its maintenance. They can be operated economically and carry less overhead charges in comparison with power machines, and are useful when the speed of production will meet the need for castings.

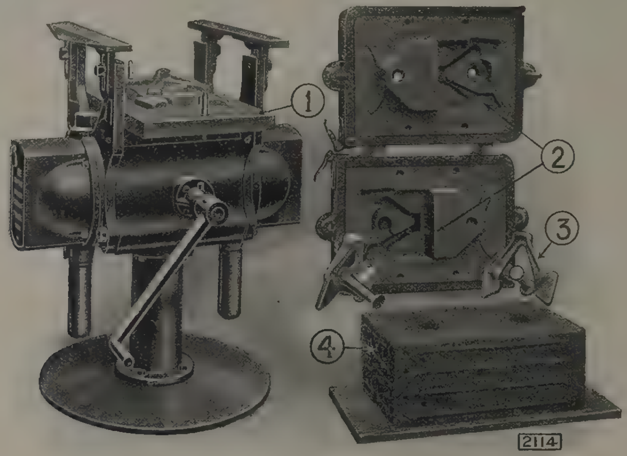


FIG. 67. HAND-RAMMING MACHINE
1. Pattern plate
2. Drag and cope
3. Casting
4. Assembled mould

These machines possess the advantages of power machines for small work, but, because more energy is expended by the operator, they are as a rule, slower.

The power machines are operated by steam, pneumatic or hydraulic pressure, and more recently machines have been designed that are operated electrically. The use of steam is not now so prevalent as formerly. Machines that are operated with this power have the disadvantage of hot pipes in the vicinity of the operator, and escapes are not only troublesome, but dangerous. Some types of machines have been demonstrated to be capable of doing efficient work under electrical power, and there seems every prospect of a profitable future for machines so operated, because the consumption of power can be localized more satisfactorily. The majority of power machines are operated by either hydraulic pressure or compressed air.

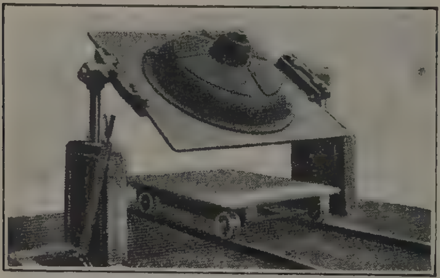


FIG. 68. HAND-RAMMING MACHINE
(The General Foundry & Engineering Co., Ltd.)

Pattern-plates. In order to obtain an increased output and great uniformity of castings of superior finish at reduced cost, much depends upon the pattern-plate. The machine may be used for ramming the mould or for extracting the pattern, or it may be designed to accomplish both operations, but, unless a first-class pattern is used, the machine is not likely to extract it without damaging the mould.

Pattern-plates are rightly recognized as labour-savers, and have an enormous economic value in foundries concerned with repetition work. They are indispensable in the making of moulds with machines. A pattern-plate consists of a plate slightly larger than the moulding box with which it is to be used, and carrying pattern sections on one or both sides from which impressions can be taken to form a complete mould when the plate has been stripped. It is necessary to provide for the registration of the moulding box on the plate. This is usually done by means of pins.

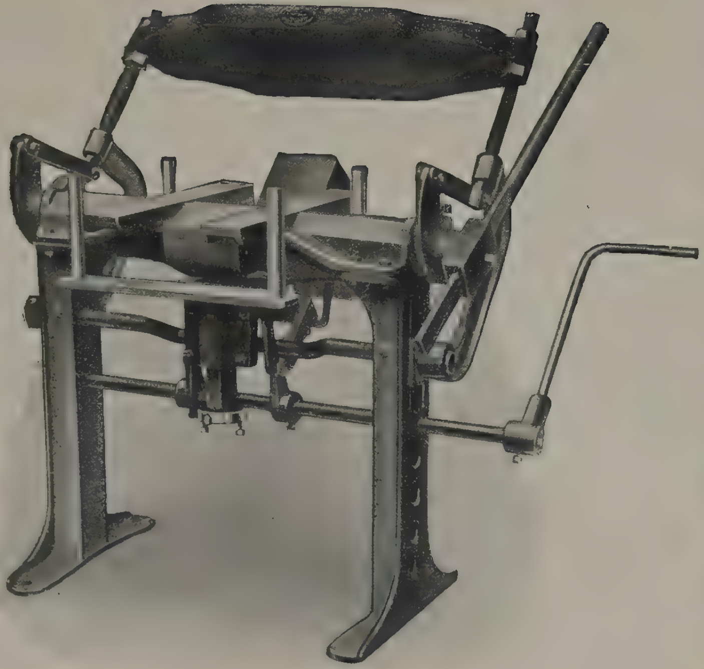


FIG. 69. "BRITANNIA" MOULDING PRESS
(Britannia Foundry Co., Ltd., Coventry)

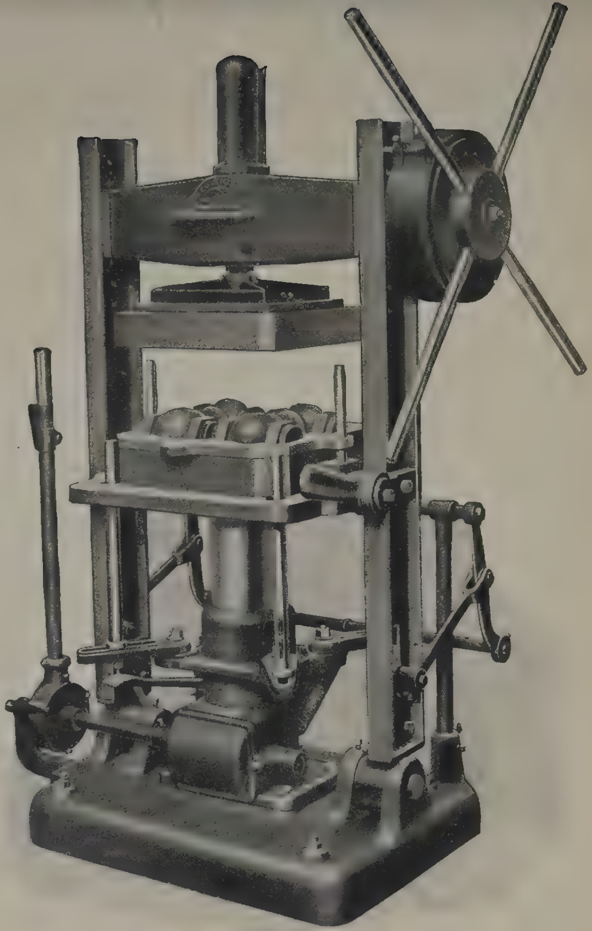


FIG. 70. THE "COVENTRY" PATENT MOULDING MACHINE
(Britannia Foundry Co., Ltd., Coventry)

Although there are many forms of pattern-plates, there are but two main types, the turnover plate and the reversible plate. The turnover plate (Fig. 64) carries pattern sections on each side, the drag being prepared from one side and the plate turned over for preparing the cope. In some instances the pattern sections are secured to separate plates for drag and cope, and separate machines are used for preparing each part of the mould. This is done to obviate turning over the plate when many moulds are to be made. The reversible plate (Fig. 65) carries the pattern sections on one side only, but they are so arranged that two impressions made will form a complete mould. In assembling the parts of the mould it is necessary to reverse them. This type of plate is very popular for work that is symmetrical, or when it is not important which side of the casting is made in the cope. Pattern-plates may be of wood, metal, plaster, or a wax composition, for hand moulding, but, for power machine work, metal or wood plates give the best service. Wood plates are used because pattern sections can be more readily secured to them, and modifications can be more easily made. The metal plate, on the other hand, is more rigid, and is more likely to retain its form. Metal plates are generally of cast iron, although aluminium is sometimes used, it being considerably lighter and not subject to oxidation to such an extent as iron. When the mould joint is flat, standard plates are used on which wood or metal patterns are fixed, and, whether a turnover or reversible plate is prepared, it is very important that the pattern sections are set accurately. The parts of the mould should complete the required shape correctly when they are assembled. Suitable runners are secured to the plates, and every effort made to dispense with tool work on the completed mould.

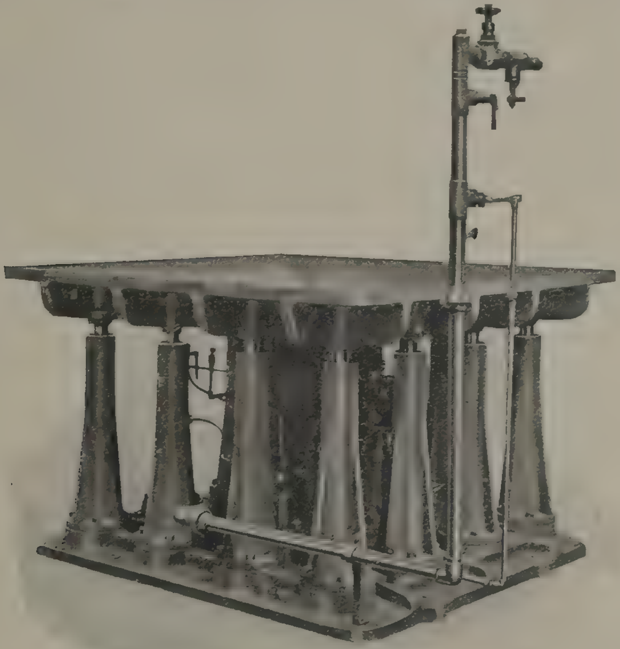


FIG. 71. JAR-RAMMING MACHINE
(Messrs. Macdonald & Son, Glasgow)

The standard flat-plate is not suitable for work involving a varying mould joint; for such work an all-cast pattern-plate is usually made.

To make such a plate a mould is prepared from a master pattern and, instead of the two parts of the mould being assembled in the ordinary way, the joint is raised to form the plate and, when assembled and cast, moulds can be reproduced from it. Either turnover or reversible plates are made in this way, but a special box-part or frame is necessary for the latter, because all the pattern sections must be cast on one side of the plate (see Fig. 66).

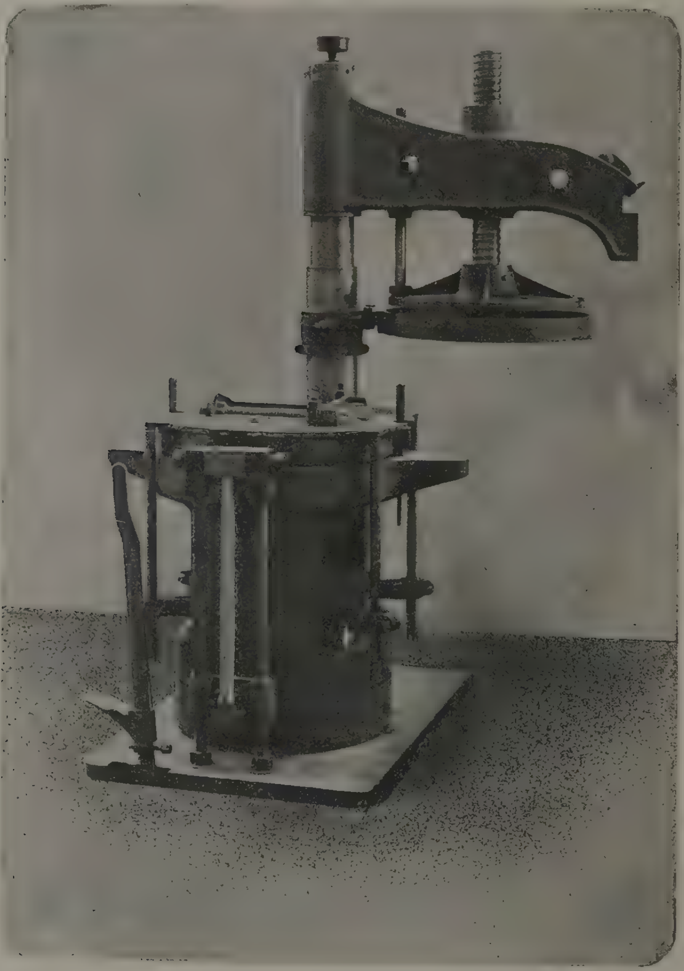


Fig. 72. Power-press Machine

Hand-ramming Machines. Although classed as a moulding machine, this type is really a stripping machine, because the manipulation of the sand is done by hand. Two representative forms of this machine are illustrated. In that shown in Fig. 67, the moulding box is supported on adjustable strips, which can be moved to accommodate different sized boxes. The pattern-plate is secured to uprights controlled by the draw handle. After ramming the mould, the pattern-plate is lowered by operating the handle.

The machine shown in Fig. 68 has its pattern-plate raised or lowered by means of long connecting rods, hinged to an intermediate part of the shafts at their upper ends, while their lower ends are connected with long cranks on a horizontal shaft.

Hand-press Machines. The hand-press or squeezer machines are quickly operated, and are effective for small or shallow work. The press on the machine in Fig. 69 is operated by the lever at the side. In this instance the pattern-plate is maintained in a fixed position, the box being lifted, when stripping, by an arrangement of levers on the spindle operated by the lower handle on the side. A more elaborate form of press machine is shown in Fig. 70. The press is mechanically geared and operated by the star-wheel at the side.

With all press machines of this type, a loose frame is used on top of the moulding box as an extension piece to enclose the quantity of sand to be pressed, and into which the press head will enter as the pressure is applied. When pressing the sand, it is necessary to maintain the pressure long enough to allow air to escape, as this makes the matrix firmer.

Jar-ramming Machines. With the press type of machine the density of sand is greatest nearest the press head, and its effectiveness is, therefore, reduced as the depth of the work increases, unless the mould is pressed at intervals, which necessitates special press heads to suit the boxes and work. This difficulty led to the use of jolt- or jar-ramming machines. The principle on which these machines operate is the jar produced by a falling mass suddenly brought to rest by impact against a stationary body. The jarring action gives the sand a greater density nearest the plate upon which the box rests, and the number of jolts determines the degree of density required. The jolts may be made by hand through the medium of a lever and cam arrangement, but hydraulic power or compressed air are more commonly used. A large machine of this type designed for ramming the sand only is shown in Fig. 71. The effect of the shock on the base is taken up by pressure cushions in the columns, and these reduce vibration and its evil effect on work near the machine. Machines of similar types have facilities for pattern stripping, and some in addition to this device, have an appliance for turning over.

Power-press Machines. These machines, operated by either pneumatic or hydraulic pressure, are not likely to be displaced as they are both rapid in operation and practically free from vibration. A machine of this type and operated by hydraulic pressure is illustrated in Fig. 72. The table, to which the pattern-plate is fixed, has a large hole in its upper surface to give clearance for loose stools used for internal stripping. The ram is operated by a valve fixed to the swinging head.

PUMPS: CONSTRUCTION AND MAINTENANCE

By Owen A. Price, M.I.Mech.E.

LESSON XI

COMPLETE CENTRIFUGAL PUMPING SYSTEM

Having studied the behaviour of a centrifugal pump by itself, we shall now consider the performance of a complete system consisting of pump, piping, and accessories.

When installed in a pipe line a centrifugal pump induces a rate of flow determined by two factors, viz. the resistance to flow in the system and the capacity of the pump against that particular resistance. Therefore, the same pump may behave differently in different pipe systems, or differently after an alteration to its own system.

Behaviour of a Complete Pumping System. In Fig. 43 a pump output curve and a pumping system are shown contiguously. The vertical lift is represented as 100, but the gross total head against the pump depends upon the size of the pipes and accessories, and also upon the quantity of water flowing. The head-quantity curve gives the whole range of output of the pump alone at the particular operating speed.

When starting such a plant it would be usual to run it up to speed with the delivery valve shut, and then open the valve until the running condition is reached. At zero delivery, the head given by the pump must exceed the static lift, flow then begins as the sluice valve is opened, and the frictional head due to this flow has to be overcome in addition to the vertical lift. This supplementary head is proportional to the square of the pipe velocity (Lesson II). Two conditions have been assumed, Case I in which the pipes and accessories are small, and Case II in which they are larger. For both cases the friction head has been set off above the upper static water-level, so that the ordinate below the particular friction

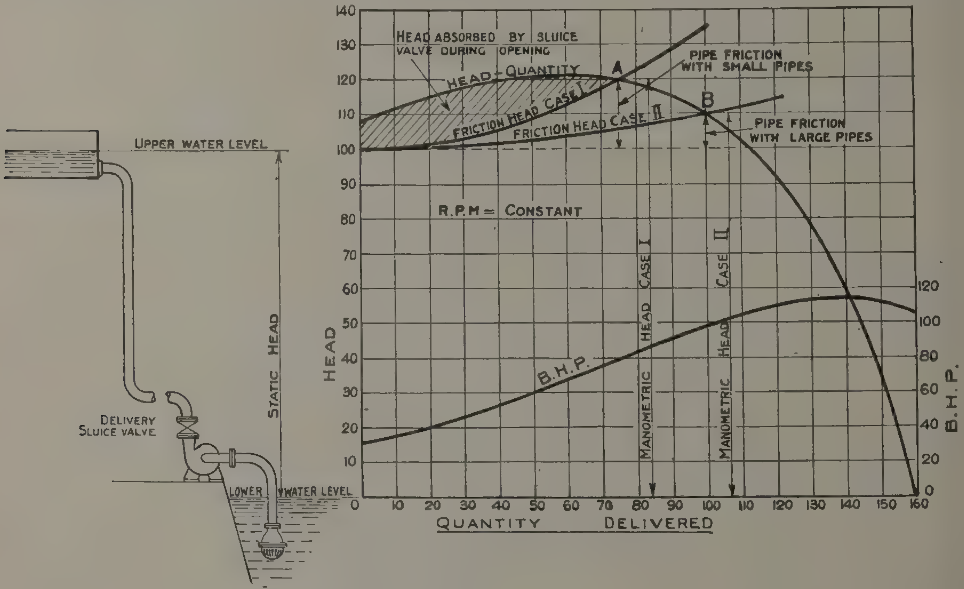


Fig. 43. Behaviour of a Complete Centrifugal Pumping System

curve represents the total head in the system at that rate of flow.

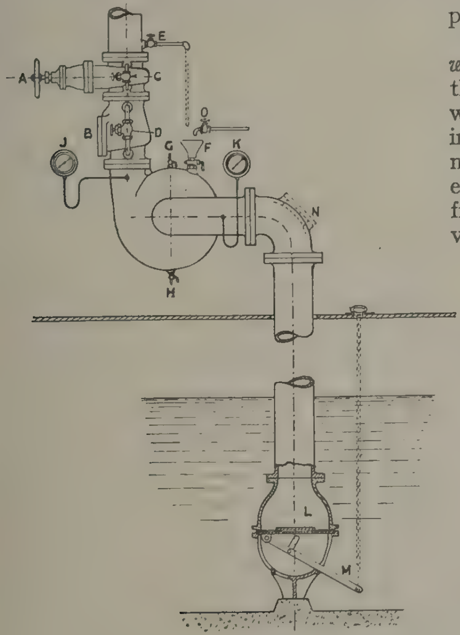
With Case I the head absorbed by the valve during opening is represented on the diagram by the ordinate intercepted between the friction curve and the pump performance curve. At *A* the sluice valve is full open and with the quantity then delivered the total head in the system is equal to the head given by the pump. Therefore *A* is the operating point of the pump for Case I and, similarly, *B* is the operating point of the same pump for Case II. For Case I the quantity delivered is only three-quarters of that for Case II, and this difference is due entirely to the use of smaller pipes and smaller or less efficient accessories.

The head against a centrifugal pump may consist chiefly of vertical lift, chiefly of pipe friction head, or of any other proportions of static and friction head, according to circumstances, but in practically all cases the amount of friction head is an important factor in determining the operating point of a complete pumping plant. Excessive friction head may mean the association of an efficient pump with an inefficient plant.

ACCESSORIES AND PRIMING ARRANGEMENTS

The equipment for a centrifugal pump, Fig. 44, is similar to that for a reciprocating pump (see Lesson VI), except that air vessels are not required. A centrifugal pump is incapable of expelling air from its casing and suction pipe, therefore some form of priming arrangement is always necessary, and special care must be taken to avoid air traps in the suction pipe.

When the pump is primed *by filling with water* a foot-valve *L* is always necessary, and this must be a freely opening type and provided with a large grating. The addition of a flushing lever *M*, to the foot-valve, is very useful in many cases. This lever may be used for emptying the suction pipe during periods of frost, or may be slightly raised for flushing the valve seat, if solid matter has become lodged there. When pumping dirty water, objects sometimes become trapped in the foot-valve, thus hindering the closing of the flap and preventing priming and starting of the pump. In such circumstances the flushing lever avoids the necessity for lifting the foot-valve.



A	SLUICE VALVE.
B	REFLUX VALVE.
C	BYE-PASS ACROSS SLUICE VALVE.
D	BYE-PASS ACROSS REFLUX VALVE.
E	DELIVERY PIPE DRAIN OR PRIMING CONNECTION
F	FILLING FUNNEL & COCK
G	AIR-RELEASE COCK
H	DRAIN COCK
J	PRESSURE GAUGE
K	VACUUM GAUGE
L	FOOT-VALVE & STRAINER
M	FLUSHING LEVER & LIFTING CHAIN
N	INSPECTION DOOR
O	FILLING PIPE WITH COCK

FIG. 44. CENTRIFUGAL PUMP EQUIPMENT

The suction pipe should always be arranged so that a convenient joint above water-level may be disconnected, if necessary, and the foot-valve lifted without difficulty.

The most suitable means for filling the pump will vary at different sites. Small pumps may be filled by hand, or by arranging a connection *O* from a water main, direct into the pump casing, or over the filling funnel *F*. If the delivery pipe is always full, a pipe *E*, with cock, may serve for priming or, alternatively, by-passes

C and *D* may be fitted across the sluice valve and reflux valve.

Where troublesome foreign matter is expected, a foot-valve should not be fitted. In such cases a large strainer and bellmouth should be provided at the pipe end, and other means adopted for charging.

Another method of priming is *by exhausting* the air from casing and suction pipe so that water is forced up to the pump by atmospheric pressure. For this purpose either a hand operated or power driven vacuum pump is provided, or an air-exhausting steam ejector or other convenient means employed. When priming *by exhaustion*, air must be prevented from rushing back through the delivery pipe by fitting a sluice valve *A* on the delivery branch, or by sealing the outlet in some way. A delivery valve also serves to control the discharge and allows of starting the pump with a very light initial torque.

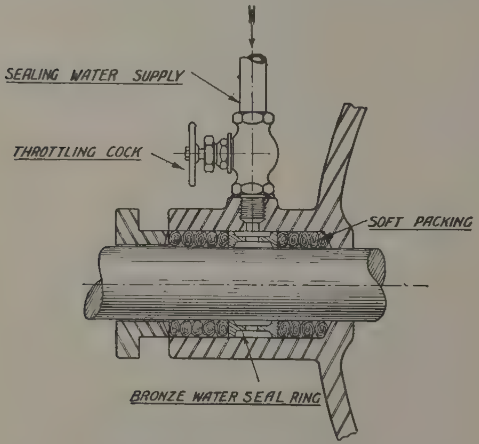


Fig. 45. Water Seal

On no account may centrifugal pumps be started until properly filled with water, because the internal bushes depend upon water lubrication and are likely to seize if the pump is run dry. With multi-stage pumps where there are many such bushes it would be disastrous to start the pump before priming.

Where the head is great, or the pipes long, a reflux valve *B* should be fitted to intercept the return shock when the pump is stopped and thus protect the pump casing.

Inspection doors *N*, on suction bends or near the impeller eye, are particularly useful when drawing from rivers, drainage sumps, etc. Grasses and stringy matter often become wound round a pump spindle, or cinders, sacking, cotton waste or pieces of wood become jammed in an impeller. The inspection door permits the removal of such obstacles, and also facilitates the necessary periodical examination of the impeller.

Stuffing Boxes. The stuffing boxes for a rapidly rotating pump spindle differ from those of a reciprocating pump (Lesson V), and are usually employed to pack against vacuum and very light pressure only. To prevent ingress of air along the pump spindle to the suction a water seal is provided, consisting of a continuous ring of water round the spindle. In the best forms a bronze water seal ring (Fig. 45) is arranged within the packing and a throttling cock fitted to the sealing water supply. The stuffing box is then filled with loosely twisted hemp, cotton or flax impregnated with tallow and flake graphite, and the gland is lightly screwed up so that a small trickle of water leaks constantly from the gland. The sealing water is usually drawn from the delivery side of the pump (see Fig. 47), but when dirty water is being pumped an external clean supply is preferable.

When pumping chemically contaminated water, which dissolves the grease in the packing, twisted hemp impregnated with white lead and graphite should be substituted for the ordinary type.

On account of the high rotary speed, all stuffing boxes should receive regular attention and the packing should be renewed periodically. For the best results the stuffing boxes and glands should be lined with gun-metal, because with certain waters the packing quickly becomes hardened by corrosion from cast-iron surfaces and is then useless and rapidly scores the spindle.

Renewable Bushes. Good modern pumps are fitted with renewable bushes for all wearing points so that the pump may be reconditioned at small cost and little inconvenience. The spindle should have gun-metal sleeves sheathing the portion exposed to the water, neck bushes should be fitted to the casing where the spindle passes through, and the impeller shoulders should run in renewable rubbing rings. On multi-stage pumps renewable bushes should be provided at every stage for both spindle and impeller.

Flexible Coupling. A centrifugal pump should be driven through a flexible coupling to prevent the transmission of either axial or side thrusts from the driving motor to the pump spindle. The pin and pad type (Fig. 46) is usually preferable, the pads being of high grade rubber lightly loaded or of leather discs for low speed heavy drives. Cheap grade rubber pads have a very short life, but pure rubber, properly bushed and on a correctly aligned coupling will last almost indefinitely.

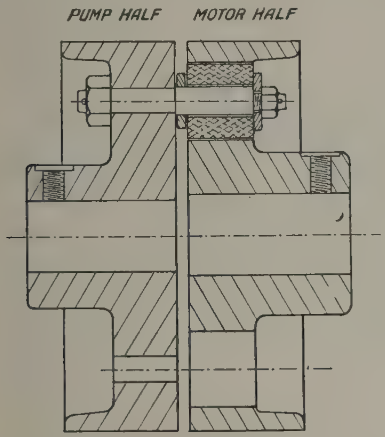


Fig. 46. Flexible Coupling

Foundation. A stiff deep section bed-plate is essential for a high speed pump. The foundation may be light but with sufficient rigidity for preservation of accurate alignment of bearings and rotors. Special care is necessary in aligning the spindle when setting the pump down to its work, and if a flexible coupling is fitted the two halves should be truly aligned. When completely erected the whole rotor should turn easily by hand or trouble will be experienced when full speed is reached.

Piping should always be supported independently of the pump, or the weight of the pipes, *especially when filled with water*, is liable to distort the pump casing and cause excessive internal wear and to throw bearings out of alignment.

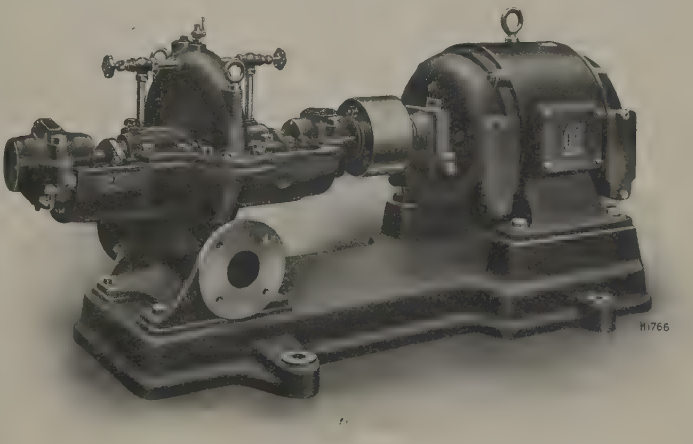


Fig. 47. Single-stage Centrifugal Pump

PIPING

The utmost importance must be attached to a simple layout of piping, pump, and accessories. All pipes and accessories should be of large bore to permit moderate water velocities, and to absorb as little friction head as possible (see Fig. 43). Bends should be of large radius and as few in number as permissible, so that circuitous piping is avoided and direct leads obtained. On the suction side of the pump this is doubly necessary, and every effort should be made to place the pump as near to the water as possible in both horizontal and vertical directions.

The maximum practical suction lift for a centrifugal pump depends upon the quality of the water, and the arrangement of the piping more than upon the pump, but the lower the suction lift the better in all cases. Pumps may work with 28 ft. vertical suction lift and 32 ft. on the vacuum gauge, but no responsible maker could recommend such practice. A slow-revolution pump can operate at higher suction lifts than a quick revolution pump, and a large pump at higher suction lift than a small one. If the water is warm, or is highly aerated or gaseous, the permissible suction lift would be much reduced. For quite small pumps 6 to 10 ft. suction lift is sufficient, 12 to 18 ft. for medium sized pumps, and 20 to 22 ft. for large slow speed pumps.

The profile of the *top* of the suction pipe should incline upwards all the way to the pump, and the number of pipe joints should be as few as possible and very carefully made. Preferably, the entire suction pipe should be tested to a pressure of 30 lb. per sq. in. to ascertain that everything is quite tight. A leakage of 1 per cent (by volume) of air may reduce a pump delivery by 20 to 25 per cent, and a slightly greater volume will break the suction column and stop the pump altogether.

The submergence of the pipe end should be 2 to 3 ft. to prevent the cores of surface eddies entering the suction pipe and admitting gulps of air, also plenty of free space should be allowed all round the pipe end.

PRACTICAL EXAMPLES

A typical modern motor-driven single-stage centrifugal pump is shown in Fig. 47. The casing is of the split type, referred to under Fig. 28E, and the pump is provided with a flexible coupling, water seals, and renewable bushes, as previously described. Such pumps are made with branches from 2 to 30 in. diameter, and are suitable for any output up to 30,000 gals. per minute against heads up to 300 ft.

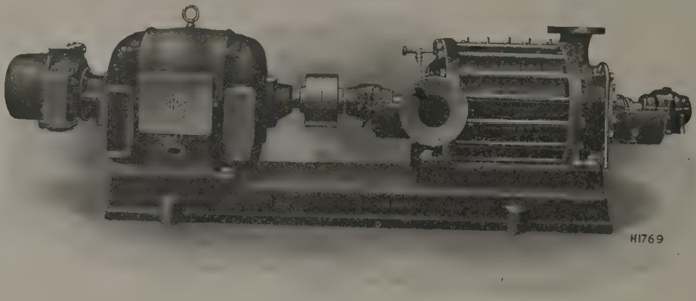


Fig. 48. Multi-stage Turbine Pump

An example of a multi-stage turbine pump with guide vanes is shown in Fig. 48. In this type the casing is built up in rings, each of which contains an impeller and guide vane and constitutes a "stage." Pumps of this class are standardized for outputs from 25 to 1,500 gals. per minute against heads up to 3,000 ft. of water.

(Conclusion)

MATHEMATICS FOR ENGINEERS

By W. G. Bickley, M.Sc.

LESSON XXIX

EXPONENTIALS AND LOGARITHMS

Growth and Decay Problems. In practically all branches of science and engineering, we meet from time to time with growth and decay problems, all characterized by the fact that the *rate* of growth or decay depends upon the actual value of the growing or decaying quantity. Sometimes (and with organic growth this is usually so) the law is complex, but very commonly it is simple—the rate is *proportional* to the quantity. Thus, if we consider the way in which the tension in a belt "grows" from the slack to the tight side, we find that, if T is the tension at any point, the angular distance of this point being θ (radians), and the coefficient of friction μ, then

$$\frac{dT}{d\theta} = \mu T.$$

Newton's law of cooling states that the rate at which a body cools is proportional to the excess of temperature above that of its surroundings, or, if θ is the temperature excess, and t is time,

$$\frac{d\theta}{dt} = -k\theta$$

where k is a constant depending upon the "emissive power" of the surface of the body. Again, when an electric current is switched off, the current does not die away instantaneously if the circuit contains inductance; if R is its resistance, and L the coefficient of self-inductance, and i the current,

$$L\frac{di}{dt} + Ri = 0, \text{ or } \frac{di}{dt} = -\frac{R}{L}i.$$

The discharge of a condenser, whether quickly through a small resistance, or slowly through a large "leak" resistance, is found to obey the law

$$\frac{dq}{dt} = -\frac{1}{RC}q$$

q being the charge at any time, t, C the capacity, and R the resistance. The chemist finds that his reactions do not occur instantaneously, but that the rate is proportional to the amount of substance still unchanged—the same law. In Compound Interest, the interest is proportional to the *amount*, which is a similar case, except that the amount grows by steps as successive instalments of interest are added year by year or quarter by quarter. On account of this similarity, the law of growth we have seen to be common, i.e. when the rate of change is proportional to the quantity growing, is often termed the "Compound Interest Law." It will be seen that in all the above instances the formula is of the type

$$\frac{dy}{dx} = ay \quad . \quad . \quad . \quad (1)$$

where a is a constant. We seek a formula for y that satisfies this equation.

Solution of the Equation. A first attempt to solve this equation would be to invert, and write

$$\frac{dx}{dy} = \frac{1}{ay}, \text{ or } a \,.\, dx = \frac{dy}{y}$$

Our endeavours to integrate the right-hand side are not successful, for this is the case in which the rule for powers breaks down, the value given by the rule being $\frac{y^0}{0}$ which appears to be infinite, and is certainly unintelligible as far as numercial calculation is concerned. We must therefore discover a different method of attack, and it will be convenient, at the same time, to take the simplest form of the equation, i.e. that when $a = 1$. We proceed to find a formula for y if

$$\frac{dy}{dx} = y \quad . \quad . \quad . \quad (2)$$

Of course, we shall need to know the value of y at the start, and it is sufficient to let y start by being 1. (The bank clerk works out his interest from tables giving that on £1 or £100, by multiplying that per pound by the amount he is calculating interest upon. Similarly, if two quantities obey the law (2), and one starts by being 3 times the other, it will grow 3 times as fast, and so will *always* be 3 times as big. By a similar argument, if we can find the formula for y in (2), starting with the value 1, we can obtain the formula if it starts with any other value, denoted by y_0, if we multiply throughout by y_0.)

Now if y starts at 1 and grows,

$$y = 1 + \,.\,.\,.\,.\,.\,.\,.\,.\,.\,.$$

the dots indicating something yet to be found, But if $\frac{dy}{dx}$ is going to be equal to y, we must also have

$$\frac{dy}{dx} = 1 + \,.\,.\,.\,.\,.\,.\,.\,.\,.\,.$$

This gives us a clue to the extra something in the y formula for the 1 in $\frac{dy}{dx}$ did not come from differentiating the 1 in y. It must therefore have come from differentiating the "something." But 1 is the result of differentiating x; there must have been an x in the "something." Hence

$$y = 1 + x + \,.\,.\,.\,.\,.\,.\,.\,.\,.$$

But $\frac{dy}{dx}$ is to be equal to y, so that

$$\frac{dy}{dx} = 1 + x + \,.\,.\,.\,.\,.\,.\,.\,.$$

Again, the x in $\frac{dy}{dx}$ could not have come by differentiating the part of y we have already found; x comes from differentiating $\frac{x^2}{2}$, so that there must be an $\frac{x^2}{2}$ in y. Thus

$$y = 1 + x + \frac{x^2}{2} + \,.\,.\,.\,.\,.\,.\,.$$

and then

$$\frac{dy}{dx} = 1 + x + \frac{x^2}{2} + \,.\,.\,.\,.\,.$$

The $\frac{x^2}{2}$ in $\frac{dy}{dx}$ necessitates a term $\frac{x^3}{2 \times 3}$ in y—and therefore in $\frac{dy}{dx}$; this in its turn necessitates a term $\frac{x^4}{2 \times 3 \times 4}$ in y, and consequently in $\frac{dy}{dx}$—and so on, for ever.

We can now collect the results, and exhibit the way in which term after term was obtained by the arrows—

$$\begin{tikzcd}
y = & 1 + \arrow[dr] & x + \arrow[dr] & \frac{x^2}{2} + \arrow[dr] & \frac{x^3}{2 \times 3} + \arrow[dr] & \frac{x^4}{2 \times 3 \times 4} + \arrow[dr] & \frac{x^5}{2 \times 3 \times 4 \times 5} + \arrow[dr] & \frac{x^6}{2 \times 3 \times 4 \times 5 \times 6} + \\
\frac{dy}{dx} = & & 1 + \arrow[u] & x + \arrow[u] & \frac{x^2}{2} + \arrow[u] & \frac{x^3}{2 \times 3} + \arrow[u] & \frac{x^4}{2 \times 3 \times 4} + \arrow[u] & \frac{x^5}{2 \times 3 \times 4 \times 5} + \arrow[u]
\end{tikzcd}$$

It is usual to complete the denominators by the factor 1 at the beginning, and they are then "factorials," as already met with in the formula of the binomial theorem. The result can then be written more concisely. We may also, in doing this, put in the factor y_0, the initial value of y if not unity, and then our result can be stated—

$$\text{if} \quad \frac{dy}{dx} = y$$

$$\text{then} \quad y = y_0\left(1 + x + \frac{x^2}{2!} + \frac{x^3}{3!} + \frac{x^4}{4!} + \frac{x^5}{5!} + \,.\,.\,.\,.\right) \quad . \quad (3)$$

where the law of the series must be evident. It may be objected that a series that goes on for ever is not much use for calculation—that life is too short to start doing a thing we shall never get to the end of—but that is not true. It must be evident that the denominators mount up rapidly, so that the terms will decrease rapidly. We shall use the series for calculation in the sequel, and it will then be clear that it *is* possible to derive numerical results from it. Our next task is to bring in the "a" we omitted in (1) to obtain the simpler (2). To do this we follow exactly the same process, but as $\frac{dy}{dx}$ is to be equal to ay, we must multiply every term of y when found by a, before putting it as a term in $\frac{dy}{dx}$. This means that for every downward arrow, we multiply by a. The process then appears as

$$\begin{tikzcd}
y = & 1 + \arrow[dr] & ax + \arrow[dr] & \frac{a^2x^2}{2} + \arrow[dr] & \frac{a^3x^3}{2 \times 3} + \arrow[dr] & \frac{a^4x^4}{2 \times 3 \times 4} + \,.\,.\,. \\
\frac{dy}{dx} = & & a + \arrow[u] & a^2x + \arrow[u] & \frac{a^3x^2}{2} + \arrow[u] & \frac{a^4x^3}{2 \times 3} + \,.\,.\,. \arrow[u]
\end{tikzcd}$$

where we have gone far enough to see that a occurs in each term of y raised to the same power as x; in other words, the series is the same series, but with (ax) in place of x. Putting in the starting value y_0 again, this second result (which, of course, includes the first) may be stated—

$$\textbf{if} \quad \frac{dy}{dx} = ay,$$

$$\textbf{then,} \quad y = y_0 \left\{ 1 + (ax) + \frac{(ax)^2}{2\,!} + \frac{(ax)^3}{3\,!} + \frac{(ax)^4}{4\,!} + \ .\ . \right\} \quad . \quad . \quad . \quad (4)$$

Now to use this series in a numerical case. Let us take that of the belt and pulley, where $\mu = 0{\cdot}2$, and the angle of lap is π (radians = 180°). Then, we know,

$$\frac{dT}{d\theta} = \mu T,$$

so that, if T_0 is the tension on the slack side,

$$T = T_0 \left(1 + (\mu\theta) + \frac{(\mu\theta)^2}{2\,!} + \frac{(\mu\theta)^3}{3\,!} + \ .\ . \right)$$

To find the tension on the tight side, we must put $\theta = \pi$, and $\mu\theta = 0{\cdot}2\pi = 0{\cdot}6284$. Thus

$$\begin{aligned} T_1 &= T_0 \left(1 + 0{\cdot}6284 + \frac{0{\cdot}6284^2}{2\,!} + \frac{0{\cdot}6284^3}{3\,!} + \ .\ .\ . \right) \\ &= T_0 (1 + 0{\cdot}6284 + 0{\cdot}1972 + 0{\cdot}0414 + 0{\cdot}0066 + 0{\cdot}0007) \\ &= 1{\cdot}874\ .\ .\ T_0. \end{aligned}$$

We see the terms diminishing rapidly, as was predicted.

It must be admitted, however, that to calculate by means of the series would be laborious, though the values might be tabulated as are logarithms, sines, etc. There is, however, another way round, and this is suggested by the way in which compound interest is calculated for long periods. Still considering $\frac{dT}{d\theta} = \mu T$, we have

$$\delta T \doteqdot \frac{dT}{d\theta}\,\delta\theta = \mu T\,.\,\delta\theta$$

whence

$$T + \delta T \doteqdot T + \mu T\,.\,\delta\theta = T(1 + \mu\,.\,\delta\theta)$$

showing that in a step $\delta\theta$, the tension is *multiplied* by the "growth factor" $(1 + \mu\,.\,\delta\theta)$. For n steps, it is multiplied by this factor n times in succession, i.e. by $(1 + \mu\,.\,\delta\theta)^n$.

Thus $\quad T_1 \doteqdot T_0 (1 + \mu\,.\,\delta\theta)^n.$

(This is, of course, approximate, but will be better and better the smaller the step.) First, take 10 steps, when $\mu\,.\,\delta\theta = \frac{0{\cdot}2 \times 3{\cdot}14}{10}\ .\ . = 0{\cdot}0628..$, and so

$$\begin{aligned} T_1 &= T_0 (1 + 0{\cdot}0628)^{10} \\ &= 1{\cdot}838\ .\ .\ T_0 \end{aligned} \quad \left| \begin{array}{l} 0{\cdot}02644 \text{ (log. } 1{\cdot}0628) \\ \quad\ \ 10 \\ \hline \ {\cdot}2644? \\ \hline\hline \end{array} \right.$$

We have used 5 figure logs., but the fifth figure in the log. is uncertain after the multiplication. If we try 100 steps, we find

$$\begin{aligned} T_1 &= T_0 (1{\cdot}00628)^{100} \\ &= 1{\cdot}845\ .\ .\ T_0 \end{aligned} \quad \left| \begin{array}{l} 0{\cdot}00266 \text{ (log. } 1{\cdot}00628) \\ \quad\ \ 100 \\ \hline 0{\cdot}266?? \\ \hline\hline \end{array} \right.$$

and we can only rely upon 3 figures in the result. We do, however, see that the result is tending towards the result obtained from the series, but log. tables will not enable us to find the limit when $\delta\theta \to 0$. The arithmetic is distinctly shorter, and if we can combine the accuracy of the series, we shall have a usable result.

To do this, we apply the "growth factor" method to equation (2), obtaining

$$\delta y = \frac{dy}{dx}\,\delta x = y\,.\,\delta x$$

$$y + \delta y = y + y\,.\,\delta x = y\,(1 + \delta x)$$

so that, for n steps,

$$y = y_0 (1 + \delta x)^n.$$

But, if we go from 0 to x in n steps, $n = \frac{x}{\delta x}$, so

$$y = y_0 (1 + \delta x)^{\frac{x}{\delta x}} = y_0 \left\{ (1 + \delta x)^{\frac{1}{\delta x}} \right\}^x \ .\ \ .\ (5)$$

We have now an expression, that in curly brackets, which contains δx only, and we must find out what happens to it as $\delta x \to 0$. The student may verify the following values—

δx	¼	0·1	0·02	0·001
$(1 + \delta x)^{\frac{1}{\delta x}}$	2·44..	2·59..	2·69..	2·71..

We still find that log. tables are insufficient to find the value to which these tend as $\delta x \to 0$, but we see that there *is* a definite number, just over 2·7, as the limit. This important number we shall denote by e (and soon we shall find its accurate value); and so we can rewrite (5)

$$y = y_0 e^x \quad . \quad . \quad . \quad (6)$$

Now compare (3) and (6), which must be the same thing, and obtain the very important result

$$e^{x} = 1 + x + \frac{x^2}{2\,!} + \frac{x^3}{3\,!} + \frac{x^3}{4\,!} + \quad . \qquad (7)$$

This series enables us to calculate *any* power of e without (necessarily) using logarithms, and e is the only number of which we *can* calculate *any* power *without* the help of logarithms. It is thus a natural base for logarithms, and we shall see this side of the usefulness of e later. At present it is more important to notice that it enables us to find e itself, by putting $x = 1$ in (7),

$$e = 1 + 1 + \frac{1}{2\,!} + \frac{1}{3\,!} + \frac{1}{4\,!} + \quad . \qquad (8)$$

Keeping 5 places of decimals, the calculation is

1 . . .	1·00000
1 . . .	1·00000
$\frac{1}{2\,!}$ . . .	·50000
$\frac{1}{3\,!}$ . . .	·16667
$\frac{1}{4\,!}$ . . .	·04167
$\frac{1}{5\,!}$ . . .	·00833
$\frac{1}{6\,!}$ . . .	·00139
$\frac{1}{7\,!}$ . . .	·00020
$\frac{1}{8\,!}$ . . .	·00002
	2·71828

i.e. we can take, for practical work, $e = 2{\cdot}718(3)$.

To return once more to our numerical example, we must have

$$\begin{aligned} T_1 &= T_0 e^{\mu\theta} = T_0 e^{\cdot 6284} \\ &= 1{\cdot}875 \;.\;.\; T_0 \end{aligned}$$

$$\begin{array}{rl} \cdot 4343 & (\text{log. } e) \\ \cdot 6284 & \\ \hline \cdot 26088 & \\ 838 & \\ 347 & \\ 17 & \\ \hline \cdot 2729 & \end{array}$$

agreeing with the original result obtained from the series.

Comparing the series in (4) with that in (7), we see that it is e^{ax}. We may thus write the result of our investigations on the equation of "growth problems."

$$\text{If} \qquad \frac{dy}{dx} = ay,$$

$$\text{Then} \qquad y = y_0 e^{ax}.$$

EXERCISE No. 44

1. Use the series to calculate to 4 places of decimals, the values of $e^{0.2}$, $e^{0.3}$, $e^{0.4}$, $e^{0.5}$, and $e^{-0.3}$. Verify that the following results of the index laws hold for your numbers: $e^{0.2} \times e^{0.3} = e^{0.5}$; $e^{0.4} = (e^{0.2})^2$; $e^{-0.3} = 1/e^{0.3}$; $e^{0.5} = \sqrt{e}$.

2. Draw graphs of e^{x} and e^{-x} from $x = -3$ to $x = 3$. Also, on a larger scale, the graph of e^{-x} from $x = 0$ to $x = 5$. (The last is the standard decay—or "die-away"—curve.)

DIFFERENTIATION OF EXPONENTIALS

It is an easy step to the rule for differentiating e^{ax}. If $y_0 = 1$, we can write the previous result backwards, thus

$$\text{if} \qquad y = e^{ax}, \text{ then } \frac{dy}{dx} = ay = ae^{ax},$$

giving us the result sought. It will be seen that to differentiate e^{ax}, we have to multiply by a—the simplest rule of differentiation there is. For instance,

$$\text{if} \qquad y = e^{3x}, \qquad \frac{dy}{dx} = 3e^{3x};$$

$$\text{if} \qquad y = e^{-4x}, \qquad \frac{dy}{dx} = -4e^{-4x};$$

$$\text{if} \qquad y = 6e^{2x}, \qquad \frac{dy}{dx} = 2 \times 6e^{2x} = 12e^{2x};$$

and so on.

We give a few examples, involving products, etc.

1. $y = 3x^2e^{x}$.

By the rule for products,

$$\frac{dy}{dx} = 6x^{2x} + 3x^2e^{x}.$$

2. $s = 5e^{-2x} \sin 3t$

$$\begin{aligned} \frac{ds}{dt} &= 5 \,.\, (-2e^{-2t}) \sin 3t + 5e^{-2t} \,.\, 3 \cos 3t \\ &= 5e^{-2t} (3 \cos 3t - 2 \sin 3t). \end{aligned}$$

DIFFERENTIATION OF LOGARITHMS

Having had x in the index, and having obtained the appropriate rule for differentiation, the next step is to find out how to differentiate a logarithm. The rule when found will be useful for its own sake, but even more as leading to a very important formula in integration.

If $y = \log_e x$, we must have, by the definition of a logarithm,

$$x = e^y$$

Consequently, by the rule of the preceding section,

$$\frac{dx}{dy} = e^y = x$$

Therefore

$$\frac{dy}{dx} = \frac{1}{x}$$

i.e. $$\text{if } \mathbf{y} = \log_e \mathbf{x}, \qquad \frac{\mathbf{dy}}{\mathbf{dx}} = \frac{1}{\mathbf{x}}$$

Note that this simple rule is only true for logarithms to the base e—we saw reasons for thinking that e is the *natural* base for logarithms, in the last lesson, and the simplicity of this rule is another reason. If we *do* want to differentiate logarithms to the base 10 (e and 10 are the only bases used outside textbooks of mathematics) we have

if $y = \log_{10} x = 0{\cdot}4343 \log_e x$

then $$\frac{dy}{dx} = 0{\cdot}4343 \times \frac{1}{x} = \frac{0{\cdot}4343}{x}$$

which is a much less simple formula.

We will use this formula, however, to find the difference, for 4 in the ordinary log. tables opposite to 5·5. Putting $y = \log_{10} x$, we shall have $\delta y = \frac{dy}{dx} \cdot \delta x = \frac{0{\cdot}4343}{x}\,\delta x = \frac{0{\cdot}4343}{5{\cdot}5} \times 0{\cdot}004 = 0{\cdot}00031$ . . agreeing with the difference in 5 figure tables.

If we have a logarithm of a function of x, such as $\log_e(x^2 + 2x + 5)$, we put $u = x^2 + 2x + 5$, so that $y = \log_e u$. Then $\frac{dy}{du} = \frac{1}{u}$ and $\frac{du}{dx} = 2x + 2$. Consequently

$$\frac{dy}{dx} = \frac{dy}{du}\frac{du}{dx} = \frac{1}{u}\frac{du}{dx} = \frac{2x + 2.}{x^2 + 2x + 5}$$

We can pick out of this example, the general rule for differentiating the logarithm of any function; it is

$$\text{if } \mathbf{y} = \log_e \mathbf{u}, \text{ then } \frac{\mathbf{dy}}{\mathbf{dx}} = \frac{1}{\mathbf{u}}\frac{\mathbf{du}}{\mathbf{dx}}$$

or in words, the differential coefficient of a logarithm is the differential coefficient of the quantity divided by the quantity. Thus, if $y = \log_e \sin\theta$, $\frac{dy}{dx} = \frac{\cos\theta}{\sin\theta} = \cot\theta$. As a particular instance of this, that will be employed "backwards" in integrating,

if $$y = \log_e (ax + b), \quad \frac{dy}{dx} = \frac{a}{ax + b}$$

INTEGRALS INVOLVING LOGARITHMS

By inverting the rule for differentiating a logarithm, just found, we derive a very important integration formula

$$\int \frac{1}{x}\,dx \text{ or } \int \frac{dx}{x} = [\log_e x]$$

EXAMPLES.

$$\int_2^5 \frac{dx}{x} = \Big[\log_e x\Big]_2^5$$
$$= \log_e 5 - \log_e 2.$$

At this point there are several possibilities. If we have a table of logarithms to base e (*natural* or *hyperbolic* or *Naperian* as they are termed), we may look up the logarithms at once, and get the result

$$1{\cdot}6094 - 0{\cdot}6931 = 0{\cdot}9163.$$

If such a table is not handy, we remember that $\log_e N = 2{\cdot}3026 \ \log_{10} N$, and get the result

$$2{\cdot}3026\,(\log_{10} 5 - \log_{10} 2) = 2{\cdot}3026\,(0{\cdot}6990 - 0{\cdot}3010)$$
$$= 2{\cdot}3026 \times 0{\cdot}3980 = 0{\cdot}9164.$$

There is yet another possibility. By the laws of logarithms, the difference of two logarithms is the logarithm of the quotient of the numbers; so

$$\log_e 5 - \log_e 2 = \log_e \frac{5}{2} = \log_e 2{\cdot}5$$

and tables give for this 0·9163.

The last of these methods has much to recommend it, especially when the quotient can be *exactly* expressed.

CONCLUSION

In the space at our disposal we have attempted to give an account of the essentials of Mathematics, as needed by the engineer, and also to show how the mathematics is applied to various practical problems. It cannot be too strongly urged that it is the *ability to apply* mathematics to practical problems that is important—

though to do this without a knowledge of mathematics is manifestly impossible. With the knowledge gained by the study of these articles, the student should go back to the mathematical portions of the earlier technical articles, and see that he understands them. If he needs further assistance, and examples of the application of mathematics to engineering problems, the author of these articles has written a little book on *Engineering Applications of Mathematics*, published by Pitman's at 5s., and he will find them there.

EXERCISE No. 45

1. Differentiate the following expressions—

$5e^{3x}$; $2e^{-6x}$; $e^{-2x}\cos 2x$; t^2e^{-2t}; $\log_e(3x+2)$; $\log_e x^3$; $\log_e \tan\theta$; $\log_e(x^2+3x+2)$; $x^3\log_e x$.

2. Writing $y = uv$, where u and v are functions of x, taking logarithms of both sides, and differentiating, prove the formula for differentiating a product; apply the same method to a quotient.

3. The speed of signalling in a submarine cable is proportional to $x^2 \log_e x$, where x is the ratio of the diameter of the core and in insulation. Find the value of x which makes the speed a maximum.

4. Given that $r = 5{\cdot}5$ is an approximation to the solution of the equation $\log_e r + \dfrac{1{\cdot}5}{r} = 2$, find a closer approximation.

5. If $\theta = e^{-0.4t}\sin\pi t$, find the smallest value of t for which the expression is a maximum.

6. Evaluate—

$$\int_2^4 \frac{dx}{x};\quad \int_{100}^{200} \frac{500dx}{x};\quad \int_{2\cdot5}^{10} \frac{5\ dx}{x-2};\quad \int_2^5 \frac{dx}{2x+3};$$

$$\int_{0\cdot5}^{1} \frac{dx}{3-2x}.$$

7. A gas expands from 2 cub. ft. to 10 cub. ft. under the law pv = constant, the initial pressure being 120 lb./sq. in. Find the work done.

8. The swept volume of an engine cylinder is 1·2 cub. ft., cut-off is at $\frac{1}{3}$ stroke, supply pressure 90 lb.-sq. in., back pressure 15 lb./sq. in. Find the work done per cycle neglecting all losses, and with hyperbolic expansion. If the diagram factor is 0·9, find the I.H.P. at 120 r.p.m. (double acting).

Answers to Exercises 42, 43, 44, and 45

(42)

1. $0{\cdot}3420$; $0{\cdot}5924$; 6; $\dfrac{\pi}{4}$; $-0{\cdot}5196$.

2. $v \doteq -\dfrac{1}{6\pi}(18\cos 2\pi t + \cos 6\pi t)$;

$$s = -\frac{1}{36\pi^2}(54\sin 2\pi t + \sin 6\pi t).$$

(43)

1. $8(2x+1)^3$; $\dfrac{x}{\sqrt{x^2-1}}$; $-\dfrac{1}{(x+2)^2}$; $-2\sin\theta\cos\theta$; $5\sin^4\theta\cos\theta$; $\sin\theta + \theta\cos\theta$; $-\operatorname{cosec}^2\theta$; $-\operatorname{cosec}\theta\cot\theta$; $-\dfrac{2}{(x+1)^2}$; $\dfrac{4x}{(x^2+1)^2}$; $\dfrac{n^2\sin\theta\cos\theta}{(1-n^2\sin^2\theta)^{1\frac{1}{2}}}$

2. $x = b/\sqrt{2}$.

3. $b = \frac{1}{2}d$.

(44)

1. 1·2214; 1·3499; 1·4918; 1·6487; 0·7408.

(45)

1. $15e^{3x}$; $-12e^{-6x}$; $-2e^{-2x}(\cos 2x + \sin 2x)$; $2te^{-2t}(1-t)$; $\dfrac{3}{3x+2}$; $\dfrac{3}{x}$; $\dfrac{1}{\sin\theta\cos\theta} = 2\operatorname{cosec} 2\theta$; $\dfrac{2x+3}{x^2+3x+2}$; $x^2(3\log_e x + 1)$

3. $x = e^{-\frac{1}{2}} = 0{\cdot}6065\ldots$

4. $r = 5{\cdot}672\ldots$

5. $t = 0{\cdot}4597$; $\theta_{max} = 0{\cdot}825$.

6. 0·6932; 346·6; 13·86; 0·3095; 0·3466.

7. 55,620 ft.-lb.

8. 8,287 ft.-lb.; 54·2.

(Conclusion)

THE TESTING OF PRIME MOVERS

By R. Royds, M.Sc., M.I.Mech.E.

LESSON VIII

THE TESTING OF INTERNAL COMBUSTION ENGINES

The term "internal combustion engine" is almost self-explanatory, and includes coal gas and producer gas engines, as well as the similar engines usually classed as oil or petrol engines. As a rule, a reliable estimate or measurement of the indicated horse-power for such engines is more difficult than in ordinary reciprocating steam-engines, but the brake horse-power can be measured with a fair degree of accuracy if a suitable form of brake is used. The readings observed during a test will depend upon the object or purpose of the test. If the object of the test is to determine the thermal efficiency under specified conditions, the amount of fuel used and the calorific value of the fuel would need to be obtained as well as the brake or indicated horse-power.

Measurement of Fuel Consumption. The method adopted for measuring the fuel would depend upon the character of the fuel. Coal gas or producer gas is usually measured by a gas meter arranged in the manner shown in Fig. 39, where *E* represents the engine cylinder and *M* the meter. The gas supply enters the meter at *A*. Between the meter and the engine the gas should pass through some form of anti-pulsator *D* to prevent excessive fluctuations of gas pressure at the meter and in the supply pipe *A*, especially if coal gas is used. The temperature of the gas can be measured by a thermometer *T* passed through a cork accommodated in a suitable *T* piece as shown. The gas pressure would be measured by a water manometer as at *G* in conjunction with a reading of the barometer. In a single cylinder four-stroke engine, especially if governed by the hit and miss method, the reading at *G* would be found to fluctuate or pulsate, and an average reading would be noted. If coal gas is used and the engine has hot tube ignition, the heating gas for the hot tube would be taken from some connection, such as shown at *C*. In important tests the meter should be tested against a standard gasometer either before or after the tests.

The calorific value of the coal gas would be determined, preferably during the engine test, by burning gas in the ordinary way in a standard form of gas calorimeter. With producer gas it would be necessary to collect gas from the supply pipe at a uniform rate during the test in a small gasometer. The calorific value of the gas would then be obtained by burning the gas in the calorimeter. Because of possible variations of gas pressure and temperature, the gas consumption and calorific value are usually corrected to the standard pressure of 30 in. barometer and temperature 60° F. by aid of the well-known gas laws.

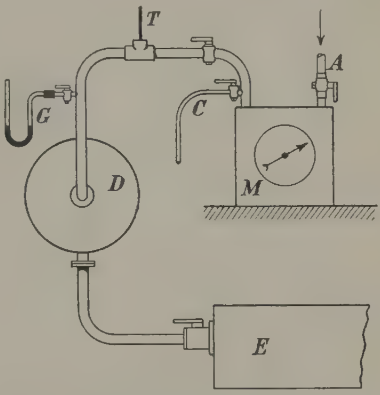


Fig. 39. Connections between Gas Meter and Engine

If the engine uses producer gas, the test might be required to include the producer in order to determine, say, the fuel consumption of the plant under definite conditions of load. To determine with reasonable accuracy the coal consumption of a gas producer is even more difficult than that of a boiler. The reason for this is the difficulty of keeping constant conditions in the producer itself, with particular respect to the amount of ashes, clinker, etc., contained in the producer. If the producer can be placed complete on a weighing machine, and connected to the scrubber through a water seal pot, the estimate of fuel consumption is more rapidly obtained. If the temperature of the producer is reasonably constant before the test begins the decrease in weight of the producer during the test represents the weight of combustible, mostly carbon, used in making the gas, supposing no fuel is added and no ashes are withdrawn during the test. In the ordinary arrangement of gas producer, however, probably the best that can be done is to keep the producer full and withdraw ashes at an agreed upon rate based on previous experience with the class of fuel used. The fire should be poked occasionally to try to ensure that no holes are left in the bed of fuel. The longer the duration of the test the less is likely to be the error resulting from changes in the quantity of fuel in the producer.

The fuel could be measured before firing into the producer by methods similar to those used in boiler tests.

Some of the gases collected for the determinations of the calorific value could be used for analysis if required. Samples of the exhaust gases from the engine could be similarly withdrawn or collected for analysis. Such an analysis of the exhaust gases shows the character of the combustion in the engine, and, provided the analysis of the fuel gas has been obtained, an estimate of the air supplied to the engine can be determined by calculation based on the analyses and the known laws of chemical combination between the oxygen of the atmosphere and the carbon and hydrogen in the gas.

The air supply to an engine may be measured directly by passing the air through a meter or through a calibrated orifice. In this arrangement a large flexible bag or anti-pulsator should be placed between the meter or orifice and the engine so as to reduce the effect of pulsations.

The most accurate method of measuring oil fuel is to weigh out a definite quantity of the oil to last, say, ten to twenty minutes on full load. The oil would be poured or drained into the fuel tank supplying the oil to the engine, taking care that all other sources of supply were effectively cut off or blanked off during the test. A gauge glass on the side of the fuel supply tank might be used to indicate the oil level unless the oil were very dark coloured and viscous, for then it might be difficult to see the level of the oil exactly when the tank is emptying. The level of the oil in the tank at the start of the test would form the standard level and the time at which the standard level was reached after each weighing would be noted. Thus, the total oil used up to any of these periods in the test would be known.

Calibrated measuring tanks might be adopted for the measurement of the oil consumption, fitted with large gauge glasses on the side to show the oil level. After switching over from the ordinary supply tank to the measuring tanks the oil level in the supply tank should be noted so as to make certain that no oil was being supplied to the engine from this tank during the test.

Similar methods would be used in petrol engines, but in that case the fuel supply tank or the measuring vessels should be fitted with a nearly air-tight lid. A similar cover should be placed at the top of each gauge glass. Tight fitting lids or covers would need a small hole in each cover to supply air as the vessels emptied.

If the thermal efficiency of the engine were required the calorific value of the oil would need to be measured. For this purpose an average sample of the oil used during the test would be obtained, and the calorific value would be determined in much the same manner as for coal. An analysis of the oil might also be required from the sample collected. If the exhaust gases were also analysed the amount of air supplied to the engine for combustion is calculable, and the exhaust gas analysis should also give some idea as to whether or not the combustion was complete.

Those who are also particularly interested in the testing of locomotives, motor-cars, refrigerators, air compressors, fans, water turbines and pumps, should refer to the book mentioned on page 1327. Reference may also be made by those interested in hydraulic power plants to the 1924 Report of the Joint Committee of the Institutions of Civil and Mechanical Engineers on "Standard Tests for Hydraulic Power Plants."

(CONCLUSION)

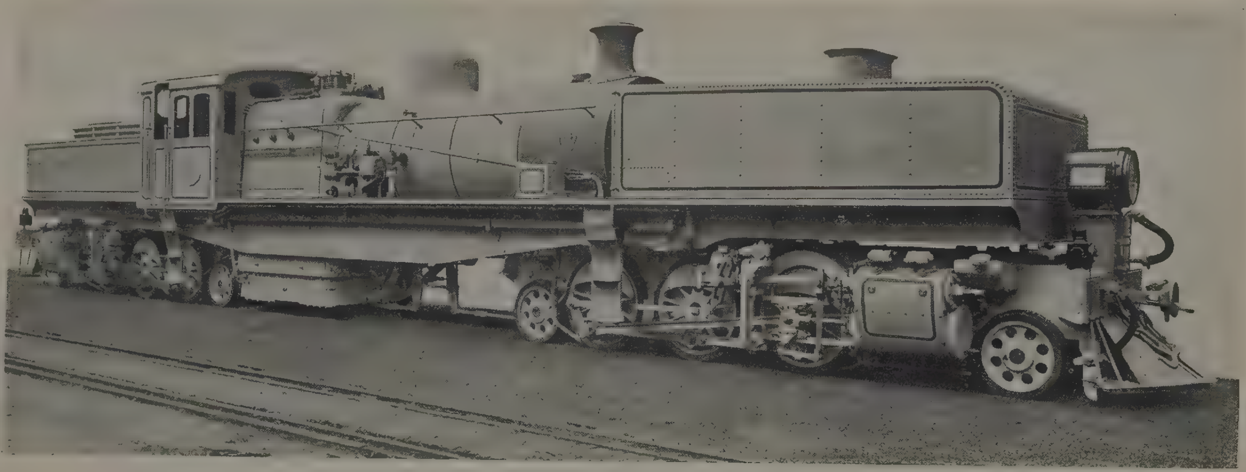


Fig. 40. South African Railways, Modified "Fairlie" Locomotive

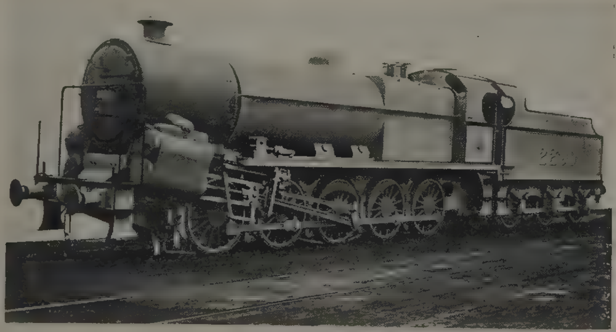


Fig. 39. L.M. & S.R.—0 10 0

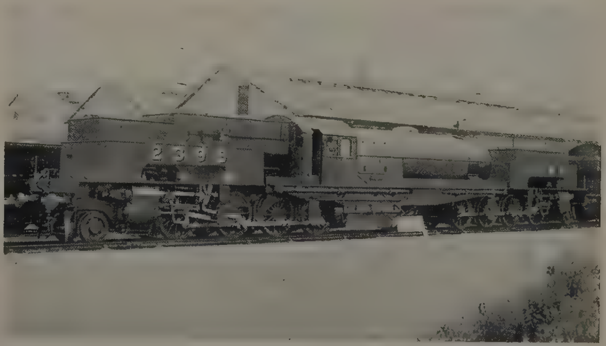


Fig. 41. L. & N.E.R. "Garratt" Locomotive

OUTLINES OF LOCOMOTIVE ENGINEERING

By A. Morton Bell, O.B.E., M.I.Mech.E., M.I.Loco.E.

LESSON X

EXCEPTIONAL TYPES OF LOCOMOTIVES

Special conditions of working, heavy grades to be negotiated, or mountain ascents to be made, have caused the adoption of many exceptional designs in steam locomotives from the very earliest days of railways. In this country there are but few sections of line requiring specialities for the haulage of trains, although recent augmentation of loads, to secure more economic operation, has caused considerable attention to be devoted to the design of larger locomotives of articulated types.

A very early example of a special engine for working an incline in this country is that of the American-built Norris engines, installed on the Lickey incline, of the Birmingham & Gloucester Railway (now part of the L.M. & S.R.), in 1840, and it will be interesting to compare an illustration (Fig. 38) of one of these, with one of their modern 0–10–0 successors of the L.M. & S.R. (Fig. 39), now operating trains on the same grade of 1 in 37.

The chief dimensions of the recent engine are as follows: cylinders (4) $16\frac{3}{4}$ in. diameter by 28 in. stroke, with ten-coupled drivers, 4 ft. $7\frac{1}{2}$ in. diameter. The tractive effort is 40,765 lb. and the locomotive is capable of performing the same work as two of the ordinary six-coupled engines previously employed in assisting trains up this incline.

The British Fairlie and the American Forney engines were both considerably adopted in their day; both relied on a flexible wheel base, secured by mounting the cylinders and driving gear on one or two self-contained bogies. The Fairlie arrangement originally consisted of a double boiler, two barrels with smoke-boxes complete, one on either side of a common fire-box. This duplex boiler was mounted on a long frame, which, in turn, was carried on two steam-driven motor bogies. The Forney was very similar; it was a double bogie engine, but only one of these was a power unit.

Some of the chief principles of the Fairlie design are still being perpetuated, and a very recent example is shown in the illustration (Fig. 40), which depicts an engine built for the South African Railways by the North British Locomotive Co. Here, however, the boiler is of normal design, resting on the main frames in conjunction with the water tanks and coal bunkers; all this superstructure is supported on two motor units, each of 2–6–2 formation.

The chief dimensions of this engine are: cylinders (4) 14 in. diameter by 23 in. stroke, with driving wheels of 3 ft. $6\frac{3}{4}$ in. diameter; the boiler carries a working pressure of 180 lb. per sq. in. The engine weighs, in working order, just under 100 tons, and has a tractive effort of 28,840 lb.

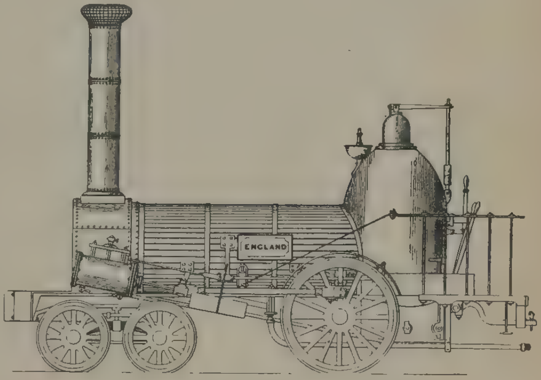


Fig. 38. The Norris Engine

All articulated locomotives must possess means of securing good, flexible, or jointed connections, for the steam and exhaust pipes, connecting up the engines with the boiler, and it is in the particular design of these details that much of the ingenuity exists. By suitably arranging the centres, or pivots, on which the driving bogies turn, a minimum movement for the steam pipes is assured, thereby securing a reduction in the sliding, or rotating action of the connections.

Other and very significant signs of the times are to be seen in the large Garratt locomotives, lately put into service on some of the British trunk lines for working heavy trains of freight and minerals over difficult sections under economical conditions. The illustration (Fig. 41) shows a 2–8–0 + 0–8–2 engine of this description, built by Messrs. Beyer, Peacock & Co., for the L. & N.E.R. The cylinders (6) are 18 in. diameter by 26 in. stroke, and the drivers are 4 ft. 8 in. diameter; the boiler pressure is 180 lb. per sq. in., the tractive effort of the engine amounting to about 62,000 lb. Although normally used for special banking services, assisting trains of 1,000 tons on an incline of 1 in 40, the successful working proves the possibility of securing much more powerful locomotives on the restricted loading gauge of the British trunk lines, if only the couplings of wagons and the employment of continuous brakes permitted them to be safely handled.

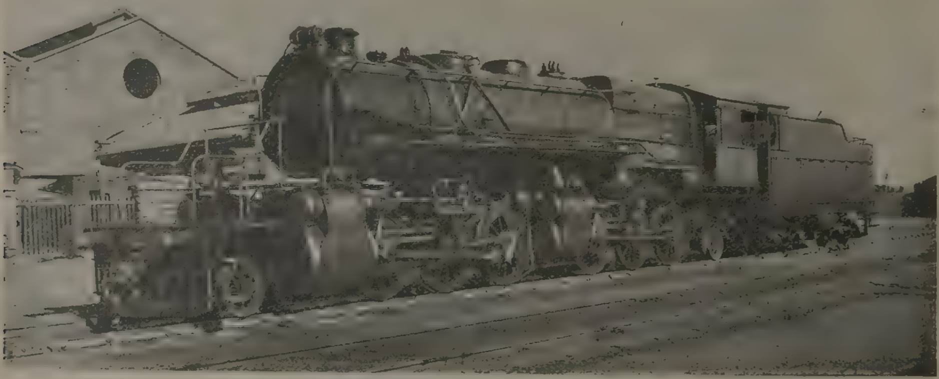


FIG. 42. N.W.R., INDIA, "MALLETT" LOCOMOTIVE

A favourite type of articulated engine which can now boast of nearly 40 years' experience is that known as the "Mallett." It is a compound with two independent groups of wheels arranged in bogies, one of which (usually the leading) is driven by the low pressure cylinders, whilst the second—rear unit—receives impulse from the high pressure cylinders. Engines of this type have reached huge proportions in the U.S.A., and there are numerous examples running on colonial and foreign railways. As an example, Fig. 42 represents an engine of this type, built for the N.W.R., of India. Constructed for the 5 ft. 6 in. gauge of track, this engine has two motor units in 2–6–0 + 0–6–2 formation. The high pressure cylinders are 19 in. diameter by 30 in. stroke, whilst the low pressure cylinders are 29½ in. diameter by 30 in. stroke. The coupled drivers are 4 ft. 4 in. diameter. The boiler pressure is 210 lb. per sq. in., and the total weight of the engine in running condition is 122 tons, the tender representing an additional 65 tons, with a full complement of coal and water; total 187 tons. The tractive effort of this effective machine is given as 52,600 lb. This locomotive is engaged in hauling trains of 640 tons over the frontier inclines of India where grades of 1 in 55 are encountered.

Although of normal outline, the unique locomotives with twelve-coupled drivers running in America and other countries must be classed among exceptional types, as clearly their construction is abnormal. An example with a wheel formation of 4–12–2, having three cylinders, 27 in. diameter by 32 in. stroke (one inside the frames, 31 in. stroke), and driving wheels 5 ft. 7 in. diameter, with a boiler pressure of 220 lb. per sq. in. The tractive effort of this huge machine is given as 96,650 lb., and it is of a class which has been put into competitive service with Mallett engines of 2–8–0 + 0–8–2 formation, and are said to be giving very excellent results.

Another interesting form of special locomotive which must not be omitted from these notes is that required for railways built up mountain sides, and relying on a rack, or toothed rail, either in supplement to, or in place of, adhesion to surmount the steep ascent. There are many forms of this type of rack, or ladder, railway, but examples in this country are few and far between.

The engine shown in Fig. 43 is a striking adaptation of a combined rack and adhesion locomotive for service on the Nilgiri Railway in South India. The engine operates over a comparatively level section of line, whereon ordinary adhesion is relied upon for traction, and then propels the train up steep grades on the mountain side, whereon a rack section is necessary to secure the haulage required. The framing carries practically two complete engines—one with two cylinders driving the wheels which are in contact with the ordinary rails, and the second operating the gear wheels which engage in the rack laid up the centre of the track. This latter is slightly higher than the rails, so that when the engine comes on to the rack section the cog-wheels automatically engage, and the steam, being admitted to the second driving gear, transfers the tractive effort to this central member of the permanent way. The dimensions of this engine are as follows: cylinders (4) 17¾ in. diameter by 16¼ in. stroke, (2) for the adhesion engine, and 16¾ in. stroke for the rack engine (2); diameter of driving wheels for adhesion, 2 ft. 8½ in. diameter; the boiler pressure is 200 lb. per sq. in.; the total weight in working order is 48 tons. The tractive effort is 30,000 lb.

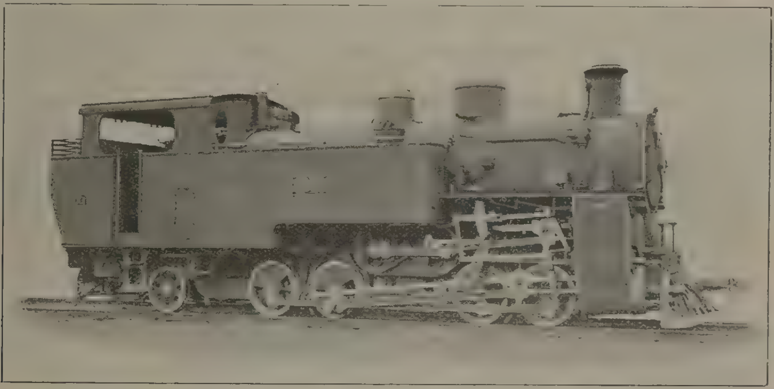


Fig. 43. Four-cylinder Compound Adhesion and Rack Locomotive for the South Indian Railway (Nilgiri Branch)

A very recent development is the adaptation of the turbine to locomotive requirements, to replace the reciprocating machine and enable, by means of a condensing system, the utilization of steam to be carried to a much higher degree. With such an arrangement, the consumption of fresh water would be reduced considerably, a very desirable attainment for countries subject to severe drought, scarcity of water, etc. The chief problem in the successful evolution of such a machine is the design of a satisfactory speed gear, whereby the elasticity of movement, so important in a locomotive, is retained, despite the fast running of the turbine motor. Electricity has been employed for the transmission of power between the prime mover, but this involves complication of apparatus, with serious loss in power, and it remains to be seen if a simple mechanical system will not eventually be found.

In the Reid-McLeod geared steam turbine condensing locomotive, shown at the British Empire Exhibition in 1924, a double bogie vehicle supports the upper structure mounted on a continuous main frame. The bogies are eight-wheeled, of which four are driving and four carrying wheels. The rear bogie is equipped with a high pressure turbine, and the front one a low pressure turbine; double reduction gear is employed. The gear is enclosed in oil-tight casings, and lubrication effected through a closed system under pressure, operated by a pump. In the driving gear, there are no reciprocating parts or coupling rods. The condenser is of the air-cooled, evaporative type, fitted with fans, for inducing air, and water vapour spraying jets. The boiler is of the normal locomotive type, with superheater and forced draft.

There are several very promising systems of turbine-driven locomotives now at work, and it seems highly probable that a satisfactory machine will be evolved in the near future. Experiments have shown that a net saving in fuel consumption of something approaching 50 per cent can be accomplished, whilst quite 25 per cent additional tractive effort is secured by the even driving power on the axles delivered by the turbine, and the great reduction in internal friction of the engine.

Whilst this treatise is concerned in locomotive engineering, it is not out of place to just mention herein the efforts being made to secure locomotives operated by the internal combustion engines in contradistinction to those relying on a steam generator for power. As, however, such machines are more or less in an experimental stage, a simple mention of the chief problems presented to the engineer engaged in solving the problem of a satisfactory arrangement is sufficient.

It is well known that an internal combustion engine gives its best result under full-load, and is anything but an economical proposition when the demands made on it for power are only a proportion of its output. To imitate the wonderful elasticity of a steam locomotive is the chief problem to be met, although there are also the difficulties of manoeuvring, etc. In the steam locomotive, exists a machine possessing a wonderful range of power. It can move at a walking pace, or assume a speed of 60 miles per hour; it can transport itself, or shift a load of maybe 1,000 tons; and all these performances will be carried out with fairly proportionate consumption of fuel and water.

In one application which cannot be passed over, viz., the Kitson-Still engine, efforts are being made to combine the advantages of the steam-engine with those of the internal combustion—a considerable proportion of the waste heat of the latter being utilized for the generation of steam for the former. It is claimed that this combination represents the most successful application of the generation of power from heat so far designed, a return of over 30 per cent being secured. Further developments on these lines are awaited with great interest.

(Conclusion)

STEAM TURBINES

By W. J. Kearton, M.Eng., A.M.I.Mech.E., A.M.I.Inst.N.A.

LESSON XII

BASIC PRINCIPLES OF STEAM TURBINE DESIGN

Introduction. The data available in the initial stages of design usually comprise (1) the power to be developed, (2) the initial pressure of the steam and superheat (if any), (3) the vacuum at inlet to the condenser, and in land work the revolutions per minute are usually known. Further, a steam consumption may also be specified and, in any case, a steam consumption must be guaranteed by the builder.

It is obvious that the number of "solutions" is infinite, although in practice, particularly in a more or less standard line, such as steam turbines of normal power coupled to alternators, the existing patterns will give some guide as to the permissible mean diameter of the blading and the number of stages. The greatest variety of designs occurs in marine work, where every job is different, and where the turbines are only part of the complete turbine-gearing-propeller system.

The calculations involved in design naturally fall under two heads, viz. (1) those relating to the determination of the principal "dimensions," i.e. mean diameter, r.p.m., number of stages, areas of nozzles, blade heights and blade angles; (2) the detail calculations referring to the strength of the blades, discs, diaphragms, and casing, the critical speed of the rotor spindle, the design of glands and so forth.

We shall endeavour, in what follows, to deal concisely with those problems which come in the first category.

Choice of Mean Diameter. This is a most important dimension. In small machines, it is decided entirely by experience, having due regard to the conflicting influence of first cost and operating costs. In larger powered turbines the mean diameter may be based on the nozzle area required for the flow of steam at the low pressure end of the turbine, reference being made simultaneously to the mean blade velocity which is involved.

Let P = S.H.P. of turbine.
,, w = weight of steam flowing through the turbine, lb. per second.
,, H_r = Rankine heat drop, B.Th.U. per lb.
,, η_o = efficiency ratio.
,, x_e = dryness fraction of the steam at exhaust.
,, v_e = volume per lb. of dry steam at exhaust
,, α = nozzle angle in last stage.
,, ρ = ratio $\dfrac{\text{blade speed}}{\text{steam speed}}$ in last stage.
,, l = radial height of last stage nozzles in inches.
,, D = mean diameter in inches.
,, N = revolutions per minute.

$$1 \text{ h.p. hour} = \frac{33{,}000 \text{ ft.-lb.}}{\text{min.}} \times \frac{60 \text{ min.}}{1} \times \frac{\text{B.Th.U.}}{778 \text{ ft.-lb.}} = 2546 \text{ B.Th.U.'s.}$$

Since the efficiency ratio is η_o, the work done per lb. of steam is $\eta_o \,.\, H_r$ B.Th.U., and the weight of steam required per second

$$= w = \frac{2546 \,.\, P}{3600 \,.\, \eta_o \,.\, H_r} \text{ lb.}$$

and since every lb. of steam occupies $x_e \,.\, v_e$ cub. ft., the total volume of steam flowing through the last stage nozzles per second

$$= w \,.\, x_e \,.\, v_e = \frac{2546 \,.\, P \,.\, x_e \,.\, v_e}{3600 \,.\, \eta_o \,.\, H_r} \text{ cub. ft.}$$

$$\text{The blade velocity} = \frac{\pi D}{12} \times \frac{N}{60}$$

$$\text{,, \quad ,, \quad ,,} = \frac{DN}{229} \text{ ft. per second.}$$

$$\therefore \text{ steam exit velocity} = V_1 = \frac{u}{\rho} = \frac{DN}{229 \,.\, \rho}$$

But the total volume of steam passing through the nozzles per second
= nozzle area in square feet × steam velocity.

$$\therefore \text{ nozzle area} = \frac{\text{volume}}{\text{velocity}}$$

$$= \frac{229 \times 2546 \,.\, P \,.\, x_e \,.\, v_e \,.\, \rho}{3600 \,.\, D.N.\eta_o.H_r}$$

The area available, expressed in terms of the diameter, nozzle angle, etc., may be set down in algebraic form by reference to Fig. 53. As already explained in Lesson VII, the nozzles are formed by casting in sheet metal vanes in the diaphragm. The area of the rectangular channel in each nozzle is $x \,.\, l$ sq. in., where $x = p \,.\, \sin \alpha - t$.

$$\text{Or, area per jet} = l \,.\, \frac{p \,.\, \sin \alpha - t}{p \,.\, \sin \alpha} \,.\, p \,.\, \sin \alpha = c \,.\, l \,.\, p \,.\, \sin \alpha$$

where $c = \dfrac{p \,.\, \sin \alpha - t}{p \,.\, \sin \alpha}$ a fraction which can be easily calculated for a standard value of p, and practical values of t and α. But the number of nozzles in the complete circumference $= \dfrac{\pi D.}{p}$

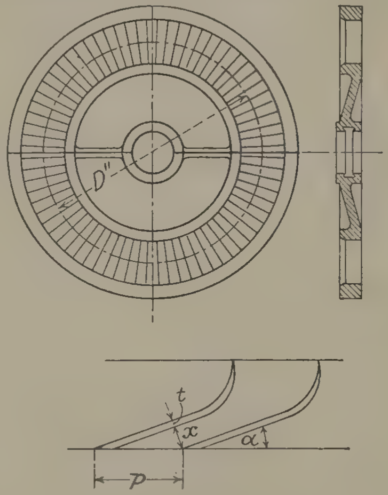


FIG. 53

$$\therefore \text{ total area} = \frac{\pi D}{p} \,.\, c \,.\, l \,.\, p \,.\, \sin \alpha \text{ sq. in.}$$

$$= \frac{\pi D \,.\, l \,.\, c \,.\, \sin \alpha}{144} \text{ sq. ft.}$$

or writing $l = m \,.\, D$, we have

$$\text{total area} = \frac{\pi \,.\, m \,.\, D^2 \,.\, c \,.\, \sin \alpha}{144}$$

Equating the area to that required, we have

$$\frac{\pi \,.\, m \,.\, D^2 \,.\, c \,.\, \sin \alpha}{144} = \frac{229 \times 2546 \,.\, P \,.\, x_e \,.\, v_e \,.\, \rho.}{3600 \,.\, D \,.\, N \,.\, \eta_o \,.\, H_r}$$

$$\text{from which } D = 19{\cdot}51 \sqrt[3]{\frac{P \,.\, x_e \,.\, v_e \,.\, \rho \,.}{N \,.\, \eta_o \,.\, H_r \,.\, m \,.\, c \,.\, \sin \alpha}}$$

The final quality x_e may be determined from the Mollier chart after making due allowance for friction losses; m is usually made about 0·2 or less, otherwise the space between the blades at the tips becomes excessive and the guidance of the steam flow tends to become imperfect. The angle α is made as small as circumstances will permit, but it may vary from 12° to 30°. Often in the preliminary stages of design, the mean diameter is provisionally fixed, having regard to a suitable blade velocity, and then the nozzle angle becomes the dimension to be found.

Number of Stages. When the mean diameter has been determined, then the "appetite for energy" of each stage may be calculated. For, knowing the diameter D and revolutions N, we can calculate the blade velocity u; this velocity, divided by ρ, gives us the exit velocity V_1 of the steam from the nozzles.

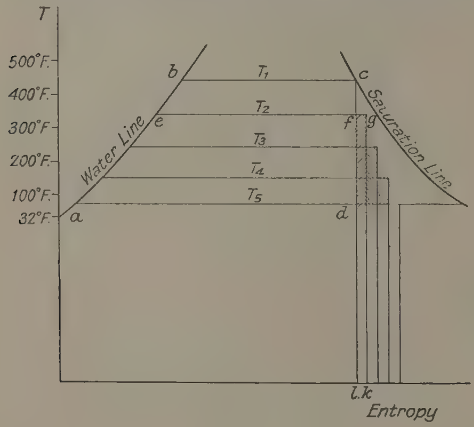


FIG. 54

Therefore the kinetic energy of the steam issuing from the nozzles is

$$\frac{V_1^2}{2gJ} \text{ B.Th.U. per lb.,}$$

and if η_n = the efficiency of the nozzles (energy basis) the heat drop per stage

$$\doteq h = \frac{V_1^2}{2g \cdot \eta_n \cdot J} \text{ B.Th.U.}$$

and the number of stages required $= \frac{H_t}{h}$ where H_t = the total heat available by expansion in the turbine.

NOTE. We have assumed above that all the stages are of the single-row pressure-compounded type. If a two-row velocity-compounded stage be incorporated in the turbine, and if its diameter be equal to that of the other stages, then it may be assumed to be equivalent, as regards "appetite," to 3 or 4 single-row wheels.

Reheat Factor. It is now necessary to show why the total heat energy available by expansion in an actual multi-stage turbine is greater than that available by adiabatic and isentropic expansion in an ideal frictionless turbine.

Consider the temperature-entropy diagram, shown in Fig. 54. Assume that T_1 is the absolute temperature of the initially dry and saturated steam, and that T_5 is the absolute temperature of the exhaust steam. Then the total energy available by adiabatic and isentropic expansion, i.e. the Rankine heat drop, H_r, is represented to scale by the area *abcd*. Let us imagine that a turbine has four stages, and let us divide the area *abcd* into four equal parts by the isothermals T_2, T_3, and T_4.

Let h_1 = heat drop in the first stage, represented by the area *ebcf*.

,, η = efficiency of each stage.

Then, the work done in the first stage is

$$\eta \cdot h_1 \text{ B.Th.U. per lb. of steam}$$

and the additional heat rejected at exhaust into the second stage nozzles

$$= (1 - \eta)\, h_1 \text{ B.Th.U. per lb.}$$

and is represented graphically by the area of the strip *fgkl*. A portion of this energy, namely, that represented by the area of the strip lying above the exhaust temperature T_5, is available for performing mechanical work in the remaining stages of the turbine. The remainder, namely

$$\frac{T_5}{T_2} \times (1 - \eta) h_1$$

is unavoidably lost.

We may look at the matter from a different point of view. Practically all the internal losses in steam turbines are friction and shock losses which "reheat" the steam and increase its dryness fraction. Now the work done on the Rankine cycle by 1 lb. of steam expanding through the second stage of the turbine between the temperature limits of T_2 and T_3

$$= (T_2 - T_3)\left(1 + \frac{x_2 L_2}{T_2}\right) - T_3 \log_e \frac{T_2}{T_3}$$

Hence an increase in the initial dryness fraction x_2 due to reheating in the previous stage produces an increase in the work done.

The same remarks apply to stages 2 and 3. The losses in stage 4 simply increase the dryness fraction of the exhaust steam. The total gain of heat drop is shown by the cross-hatching, and if H_t be the gross heat drop in all the stages, then the ratio

$\frac{H_t}{H_r}$ is known as the *reheat factor*, and is represented by the symbol R. In a fairly efficient steam turbine R would be about 1·05.

It must not be imagined that the performance of the actual turbine is in any way superior to that of the ideal turbine. It is, on the other hand, only due to the imperfections of the actual machine that the slight increase of dryness fraction resulting therefrom increases the energy available in the later stages of the turbine. It is, as it were, a sort of compensation for losses suffered and the meaning of the term *reheat factor* is very well expressed in the German equivalent *Rückgewinnungszahl*, which means literally, " winning back coefficient."

Nozzle Areas. It might be said, as a general rule, that if the nozzle areas required in the first and last stages of the turbine are calculated accurately, then no refinements of calculation are necessary in determining the areas in the intermediate stages.

The calculation of the first stage nozzle area is particularly important if the first stage of the turbine is a two-row velocity-compounded wheel, for then the final steam pressure will usually be lower than the critical, and the weight of steam which the nozzles are capable of passing is definitely limited by the throat area of the nozzles. Two cases arise for consideration.

Case I. Steam initially dry and saturated.

Let w = weight of steam per second.
,, A_t = throat area of nozzle in square inches.
,, p_1 = initial pressure, lb. per square inch abs.
,, v_1 = initial specific volume, cubic feet per lb.

Then, assuming adiabatic and isentropic expansion, it may be shown that

$$w = 0{\cdot}3003 \, . \, A_t \, . \sqrt{\frac{p_1}{v_1}}$$

It would appear reasonable to expect that the weight of steam discharged in practice would be appreciably less than that given by the above formula, owing to the effects of friction losses. There is, however, another effect, namely, *delayed condensation*. Space restrictions deny a detailed explanation, but it may be said briefly that delayed condensation tends to increase the mass flow for a given throat area. The two effects mentioned just about annul each other, and the actual discharge is given approximately by the formula.

Case II. Steam initially superheated and remaining superheated down to the critical pressure.

On the assumption of adiabatic and isentropic expansion, the mass flow per second is given by the expression

$$w = 0{\cdot}3155 \, . \, A_t \, . \sqrt{\frac{p_1}{v_1}}$$

Here, however, to be sure of a satisfactory result, it is imperative to consider friction losses. Probably the most reliable data on the efficiency of steam nozzles are provided in the various Reports of the Institution of Mechanical Engineers Steam Nozzle Research Committee. A résumé of these is presented in the writer's book, *Steam Turbine Theory and Practice* (second edition).

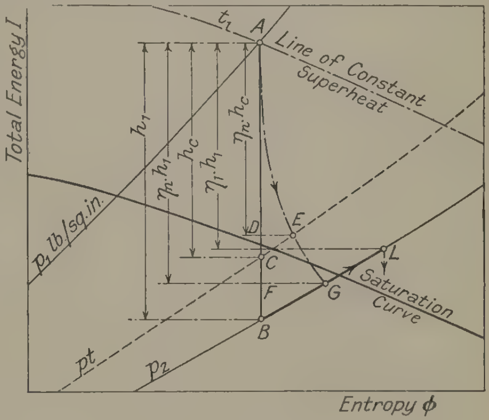


FIG. 55

In Fig. 55, suppose that A is the initial " state point " of the steam, i.e. its pressure is p_1 lb. per sq. in. absolute, and its superheat t_1° F. Let p_2 be the final pressure of the steam, and suppose that this is lower than the " critical" pressure p_c.

$p_c = 0{\cdot}5457 \, p_1$ for steam expanding adiabatically and isentropically in the superheat region. The exact value of the ratio $\frac{p_c}{p_1}$ will depend, of course, on the magnitude of the friction losses, but the value 0·55 is near enough for practical purposes.

The adiabatic AB through A will intersect the critical or throat pressure line p_c in C. Then h_c is the energy available down to the throat, and if η_n be the assumed value of the nozzle efficiency, then the total energy per pound of steam at the throat must be equal to

(initial energy) – (heat energy converted into energy of motion)

$$= I_A - \eta_n \cdot h_c = I_D = I_E.$$

Therefore E is the state point of the steam at the throat. The superheat at E may be read off the Mollier chart directly, and the corresponding specific volume v_E may be obtained from steam tables. Usually some interpolation is necessary.

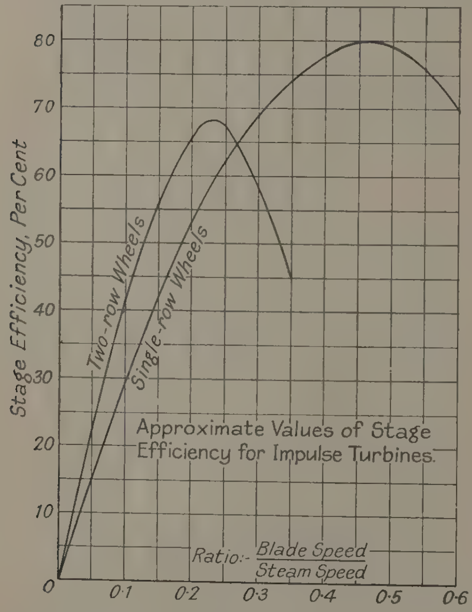


FIG. 56

The velocity V_E at the throat of the nozzle may be calculated by means of the expression

$$V_E = 223{\cdot}8\sqrt{I_A - I_E}$$
$$= 223{\cdot}8\sqrt{\eta_n \cdot h_c} \text{ ft. per sec.}$$

If the weight of steam flowing per second is w lb., then the throat area

$$A_t = \frac{w \cdot v_E}{V_E} \text{ sq. ft., or } \frac{144 \cdot w \cdot v_E}{V_E} \text{ sq. in.}$$

The total adiabatic heat drop in the nozzles is h_1, $\eta_n \cdot h_1$ represents the heat converted into kinetic energy and G is the final state point. The calculation of the outlet area is carried out in the manner described above.

It is now necessary to explain how to obtain the initial state point of the steam for the second stage of the turbine.

From a knowledge of the ratio of blade speed to steam speed in the first stage we may obtain the approximate efficiency of that stage from the curves given in Fig. 56. Let η_1 be the efficiency. Then, since the initial energy of the steam entering the second stage is equal to

$$I_A - (\text{work done in the first stage})$$
$$= I_A - \eta_1 \cdot h_1$$
$$= I_L,$$

then the point L on the p_2 line must represent the required initial condition of the steam.

The areas required in the other stages of the turbine may be obtained by a repetition of this process. The nozzle angles in the various stages are then adjusted so as to give a progressive increase of the nozzle height from the first to the last stage.

Blade Angles. The inlet angle of the blade, i.e. the angle made by the leading face of the blade on its inlet side with respect to the plane of rotation, is made approximately equal to the angle given by the triangle of relative velocities. In constant speed machines, in order to operate efficiently at partial loads, the actual inlet angle is made appreciably larger than the theoretical full-load angle.

The outlet angle is calculated with reference to the area required for the flow of steam from the blades. It is usually rather smaller than the inlet angle.

Reaction Turbines. The method of design very briefly outlined above would have to be modified somewhat before application to the impulse-reaction turbine, owing to the very great amount of calculation involved by the large number of stages. It is usual, in this case, to treat the stages in groups or "expansions" consisting of blades of the same height, the conditions as regards steam density and velocity at about the middle of the group being assumed to be the same for all elements in the group.

(CONCLUSION)

FUEL CALORIMETERS

By Professor G. E. Scholes, M.B.E., M.Sc., M.I.A.E., M.I.Mech.E.

PART II

CALORIMETERS FOR SOLID OR LIQUID FUELS

In most calorimeters designed for testing the calorific value of solid or liquid fuels, the fuel is burned in an atmosphere of oxygen, the oxygen remaining either at constant pressure or constant volume during the combustion. In one type of calorimeter the oxygen necessary for the combustion of the fuel is supplied in the form of sodium peroxide, which is mixed in a fixed proportion with the fuel. Arrangements are made for fusing a fine iron or platinum wire in contact with the sample of fuel and the sodium peroxide, and, once started, the combustion proceeds vigorously. In all cases the heat due to the combustion is transmitted to a known weight of water in which the calorimeter is submerged during the test. The weight of water multiplied by the rise in temperature gives the heat absorbed, and that is due to the combustion of the fuel. A correction for radiation has to be applied if the result is to be accurate.

Constant Pressure Calorimeters. In this type of calorimeter the weighed fuel is held in a crucible contained in a metal base supplied with a detachable glass vessel. Oxygen from a cylinder passes through a fine control valve to the top of the glass vessel, leaving at the bottom through a series of holes in the base and then bubbling up through the surrounding water.

The combustion is started by fusing a short length of fine wire in contact with the fuel by passing a current of electricity through it. The hot products of combustion are then driven out through the holes in the base of the vessel by the incoming oxygen, and give up their heat to the water in passing upwards through it.

Whilst simple apparatus of the types thus briefly described may be used to obtain calorific values of solid fuels with some degree of consistency, the results are comparative rather than absolute, and the standard instrument used is the bomb calorimeter.

BOMB CALORIMETERS

Where accurate determinations of the calorific values of solid and liquid fuels are required, an instrument called a bomb calorimeter is used.

The original form of this instrument was due to M. Berthelot.

The bomb itself usually consists of a stout hollow metal cylinder closed at one end and having a detachable cover at the other end, and is designed so that it may be filled with oxygen at a high pressure (generally about 25 atmospheres), so as to enable the fuel to be burned quickly and completely. The combustion takes place at constant volume since the bomb is gas tight, and during the test the whole of the bomb is submerged in a known weight of water. Since the weight of fuel burned, the weight of water used, and the rise of temperature of the water during combustion of the fuel are determined, the calorific value of the fuel may be calculated.

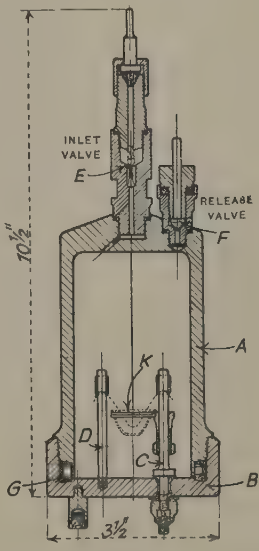


Fig. 2. Section of Bomb Calorimeter

In practice, account has to be taken of the water value of the calorimeter and the amount of heat radiated from the apparatus during the test, whilst for very accurate work the heat liberated from the combustion of the ignition wire and from the formation of small quantities of nitric acid would have to be determined. There are several modifications of the original Berthelot bomb in use, and a description of one will illustrate this type of instrument.

In the bomb shown in Fig. 2 and developed by the author, the material used throughout, with very minor exceptions, is stainless steel, which has the dual advantage of strength and resistance to corrosion. The bomb body *A*, cover *B*, and most of the details are machined from the solid. The bomb is inverted so that the cover forms the base on which it stands, and the ignition rods *C* and *D* are attached to the base. The upper part of the bomb is thus in one piece, the head of which carries the charging and discharging valves *E* and *F* respectively, the complete bomb consisting of two parts only.

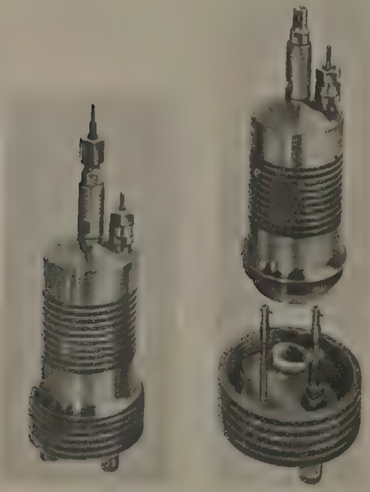


FIG. 3. BOMB CALORIMETER

Left: Bomb assembled for test
Right: Upper half raised showing crucible and coal sample

The bomb is sealed against the escape of oxygen or products of combustion by the use of a self-sealing rubber ring *G*. In other bombs a lead ring is often used as a jointing ring being held between the cover and the bomb body, the cover being attached to the body by means of studs or similar screw device. In the bomb illustrated, the rubber jointing ring fits in a chamber formed by a slight recess in the base and the under face of the body of the bomb. The inner face of the bomb body projects downwards in the form of a spigot and so forms a ring chamber. Water placed in the base of the bomb to absorb any nitric or sulphuric acid fumes thus isolates the rubber ring from the bomb body in which combustion takes place, and thus the ring is not affected in any way.

The sealing effect is produced by the pressure of the contained gas which forces the lips of the ring against the metal seats. This enables the base or cover of the bomb to be screwed up by hand, and makes the instrument easy to handle.

The oxygen enters by a non-return valve *E*, and the products are discharged at the end of the test through the screw-down valve *F*.

The crucible containing the sample is held in a cradle *K*, attached to a split sleeve, which slides along and rotates about the ignition rod *C*. The ignition wire which is usually very fine nichrome or platinum wire (diameter 0·004 in.) is held in slots, cut in the ignition rods, by means of chamfered sleeves.

The ignition rod *D* is screwed into the base of the bomb, whilst the rod *C* passes through the base and is insulated from it by means of fibre cones and mica washers. This rod *C* has a spring foot which makes contact with any surface on which the bomb stands.

The bomb, when assembled and charged with oxygen to a pressure of 25 atmospheres approximately, is placed in a copper vessel or calorimeter containing a known weight of water (generally about 5 lb.). The water used is sufficient to submerge the bomb. This calorimeter is placed inside a second and larger one, being insulated from it by means of a triangular ebonite base which centralizes the smaller calorimeter inside the larger one. This larger vessel in turn stands in a water-jacketed calorimeter, the object of this arrangement being to prevent, as far as possible, heat being radiated from or to the bomb and inner calorimeter during the test.

The outer vessel carries a clamp for holding the Beckmann thermometer and the stirrer mechanism. The stirrer mechanism, when in use, causes a stirrer to agitate the water round the bomb, and so ensures an uniform temperature in the water as well as expediting the passage of heat from the bomb to the water during the test. Papier mâché covers are placed over the calorimeter vessels to reduce air circulation and loss of heat due to this.

Method of Carrying Out a Test. In making

ERRATA

Vol. I

Page 4, column 1, line 13. For 132·4 sq. in. read 73·5 sq. in.

Page 4, column 1, line 18. For 132·4 read 73·5.

Page 120, column 2, line 7. For $CO_2 + O = 2CO$ read $CO_2 + C = 2CO$.

Page 157, column 2, Answers to Exercise 5, bottom of page, 1(*e*). For 88,570 read 11,570.

Page 234, column 2, line 28. For $\frac{(T - 2{\cdot}314p)}{\pi}$ read $\frac{(T - 2{\cdot}314)}{\pi} \cdot p$.

Page 235, column 2, line 27. For "pheripheral" read "peripheral."

Page 236, column 1, line 7. For $F = S.F.Y.p$, read $F = S.f.Y.p$.

Page 248, column 2, line 4. For 1861, read 1681.

Page 259, column 2, line 6.

$$\text{For } W = \frac{7{\cdot}728r^4}{EL^2}, \text{ read } W = \frac{EL^2}{7{\cdot}728r^4}.$$

Page 259, column 2, line 7.

$$\text{For } r = \sqrt[4]{\frac{7{\cdot}728W}{EL^2}}, \text{ read } r = \sqrt[4]{\frac{EL^2}{7{\cdot}728W}}.$$

Page 430, column 2, line 7.

For $Z_n + H_2SO_4 = Z_{n4}\,SO + H_2$,
read $Z_n + H_2SO_4 = Z_n.SO_4 + H_2$.

Page 431, column 1 (column 5 of table). For (NH_4C) read (NH_4Cl).

(5462)

Page 435, column 1, line 42.

$$\text{For } d_e = \frac{t}{p}, \text{ read } d_e = \frac{t}{P}.$$

Page 435, column 2, line 22.

$$\text{For } R = \frac{\cos \sigma_W}{\cos \sigma_p}, \text{ etc.}:$$

$$\text{read } R = Q \cdot \frac{\cos \sigma_W}{\cos \sigma_p} = Q \cdot \frac{\cos (\Sigma - \sigma_p)}{\cos \sigma_p}$$
$$= Q \cdot (\cos \Sigma + \sin \Sigma \cdot \tan \sigma_p).$$

Page 436, column 2, last line.

$$\text{For } d_e = \frac{t}{p}, \text{ read } d_e = \frac{t}{P}.$$

Vol. II

Page 542, column 2, line 24. Read "l = length of beam."

Page 547, column 1, line next to last. For $a(x - \alpha)\,(\rho c + \beta)$ read $a(x - \alpha)\,(x - \beta)$.

Page 577, column 2, line 8. For "left-hand covers" read "outer covers."

Page 691, in table at bottom of page. For "f_1" read "f_c."

Page 726, column 2, line 24. For

$$\frac{T \times t}{2} \text{ read } \frac{T + t}{2}.$$

Vol. III

Page 979, column 1, line 33.

$$\text{For } \tan \tfrac{1}{2}(B - C) = \frac{b + c}{b - c} \cdot \tan \tfrac{1}{2}(B + C)$$

$$\text{read } \tan \tfrac{1}{2}(B - C) = \frac{b - c}{b + c} \cdot \tan \tfrac{1}{2}(B + C).$$

a test a sample of the fuel is weighed out into a clean dry crucible. The weight of the sample varies with the fuel, about a gramme being used for coal and from 0·7 to 0·8 of a gramme with oil. The crucible used may be of platinum, silica ware, porcelain, or fire clay.

The crucible is placed in the bomb, ignition wire fixed, the cradle supporting the crucible and sample is adjusted so that the ignition wire makes good contact with the fuel sample, 15 cub. cm. of water placed in the bottom of the bomb, and the bomb assembled. The bomb is then coupled up to the oxygen cylinder and charged to a pressure of 25 atmospheres.

Water at a temperature of one to two degrees below the room temperature is accurately weighed or measured out into the calorimeter, and the whole apparatus assembled with stirrer and thermometer in place. The stirrer mechanism is started and, after an interval of two minutes, readings on the thermometer are taken every minute. it being essential that the increments of temperature should be approximately equal per unit of time.

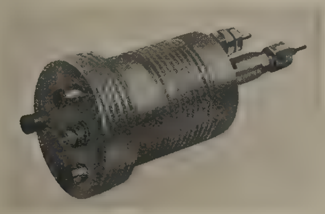


FIG. 4. VIEW OF UNDERSIDE OF BOMB

Showing insulated supports and spring contact

These readings are observed and recorded until the rise in temperature is uniform, and then the ignition switch is closed, at the next minute, a temperature reading being taken simultaneously. When the switch is closed, the ignition wire is fused and the sample commences to burn. The reading of the thermometer rises rapidly, and only approximate readings can be obtained for three or four minutes. When the rate of rise slows down accurate readings are again taken until a maximum temperature is recorded, after which further readings are taken until a steady fall is registered. The test is then completed, and the apparatus can be taken adrift and prepared for the next test. The temperature readings fall into three divisions, viz., those taken prior to ignition, during the consequent rise, and the consequent fall. The information obtained gives the rate of rise of temperature prior to ignition, the increase of temperature following ignition, and the rate of fall of temperature after the combustion has finished. The correction for radiation is calculated from the rate of rise prior to ignition, and the rate of fall at the end of the test by using the Regnault-Stohmann-Pfaundler formula which has been found sufficiently accurate for all practical purposes.

Let n = number of minutes between the ignition and the attainment of the maximum temperature.

v^1 = rate of temperature fall in degrees per minute at the end of the test.

v = rate of temperature rise in degrees per minute at the beginning of the test.

Then the radiation correction in degrees

$$= n\,v^1 + \frac{(-v + v^1)}{2}$$

and this correction is added to the rise in temperature after ignition in order to correct for the radiation effect.

EXAMPLE OF TEST.

Date of test	10th Jan., 1927
Fuel tested	Boiler fuel oil
Weight of crucible and fuel sample	2·9456 grm.
Weight of crucible	2·1380 ,,
Weight of fuel	·8076 ,,
Weight of water in calorimeter	2,268 grm.
Water value of apparatus	412 ,,
Total equivalent weight of water	2,680 ,,

Time and temperature readings—

Time (mins.)	*Thermometer Readings* (° C.)
0	0·788
1	0·790
2	0·791
3	0·793
4	0·794 (sample ignited)
5	3·150
6	3·800
7	3·856
8	3·870
9	3·871 (max. temp.)
10	3·868
11	3·862
12	3·855
13	3·849
14	3·843

Radiation correction—

From the results, $n = 5$, $v^1 = \frac{\cdot 025}{4} = \cdot 00625$ taken from the 10th to the 14th minute, $v = \frac{\cdot 006}{4} = \cdot 0015$

taken from zero to the 4th minute. Hence, radiation correction $= 5 \times {\cdot}00625 + \frac{(-{\cdot}0015 + {\cdot}00625)}{2} =$ 0·0336.

Rise in temperature during test 3·871 − ·794	= 3·077
Radiation correction	0·0336
Corrected temperature rise	3·1106
Heat absorbed by water	= 2680 × 3·1106 = 8336·4 calories
Hence calorific value of fuel tested	$= \frac{8336{\cdot}4}{{\cdot}8076}$ = 10,322 calories per grm.

This result is equivalent to 10,322 C.H.U per lb., or 10,322 × 1·8 = 18,580 B.Th.U.'s. per lb.

Water Value of Apparatus. The apparatus absorbs heat when its temperature is raised, and it is necessary to know what mass of water would absorb the same amount of heat for the same temperature rise. This mass of water is known as the water value of the apparatus. A common method of determining the water value of a bomb calorimeter is to measure the heat given to the water during the burning of a given weight of a fuel whose calorific value is known. Under such circumstances the heat given up by the fuel and that absorbed by the water are known and the difference represents the heat absorbed by the apparatus for the corrected temperature rise. If this quantity is divided by the corrected temperature rise the result is the water value of the apparatus.

NOTES ON THE PREPARATION AND TESTING OF SOLID AND LIQUID FUELS

Liquid fuels such as oil differ from solid fuels such as coal in that they are uniform in character, so that variations in the composition of different samples taken from the same lot are negligibly small. Repeated tests will thus return the same calorific value within narrow limits, any differences being due to slight errors in weighing, etc. With coal or other fuels in which considerable variation may occur it is essential that the original sampling from the bulk should be carefully done, and that the sample should be finely ground before any is weighed out for testing.

After grinding, the coal should be dried in a water oven and, as dry powdered coal absorbs moisture the subsequent weighing operations should be carried out quickly. If an independent determination is being made of the moisture it is not necessary to dry the coal, a correction for the moisture contained being easily made.

In testing fuel oil, two alternatives may be used. In the first the crucible is partly filled with dry china clay or fuller's earth and weighed. The oil is then poured into the crucible and is absorbed by the clay. The remainder of the operation is normal.

In the second no absorbent is used, and the oil is poured directly into the crucible. The latter method is simpler and has given no trouble.

With petrol or light oils the rate of burning at the moment of ignition is such as to cause a difference of pressure above and below the crucible, and if the base of the crucible is not supported it may be blown out.

With such fuels the crucible should be supported by a heavier metal crucible held in the cradle.

If the sample is weighed out and transferred to the bomb quickly the loss by evaporation is negligible, and it is not necessary to use a glass or other capsule to contain the petrol sample as is frequently done.

PATTERN-MAKING AND FOUNDRY WORK

By J. McLachlan and C. A. Otto

LESSON XIV

SETTING OUT AND DETERMINING THE CONSTRUCTION OF A LARGE PATTERN

In all pattern-shops there is a number of drawing boards of various sizes. The preliminary examination should determine the number of views that it will be necessary to set out. For some large work of plain design only a plan or elevation is necessary, in other cases two views are advisable, while in some cases it may be necessary to set out an elevation, a plan, an inverted plan, end views and sections. Cores have much to do with the number of views necessary, and it very often happens on a large job that many scrap views and sections may have to be set out as the work proceeds. It is almost impossible, say, in the case of a large cylinder, to get all the necessary views on one drawing board.

Having determined upon the number of views that will be necessary, the size of the drawing board can be decided. To set out the various views of a large pattern in their correct relative positions, as on the blue print or tracing, is seldom practicable. In many cases the drawing board has to be sent to the foundry with the pattern, as an aid to the moulder when setting cores and testing thicknesses, therefore it must be of reasonable proportions. A comparatively small board, therefore, would be used, and the various views would be drawn on top of each other and metal thicknesses coloured to avoid confusion.

Trammels are of all sizes and sorts. Often the trammel heads are either exceptionally heavy with a very light stick, or the stick is too heavy for the heads. It is quite impossible to make an accurate drawing with a trammel stick slim in proportion to its length. There should be no yield. Trammel points are invariably too long. Some craftsmen point both heads, but it is much better, even with small trammels, to give one head a chisel edge; not a wide chisel edge, but just sufficiently so that it will cut and not tear the surface of the board. A sharp point follows the grain of the timber and a chisel edge is not as liable to do so. It is possible to have the edge too sharp. A trammel so sharpened that the line drawn by it cannot be seen is useless.

What is true of trammel points is also true of compasses. With regard to marking knives, some prefer the knife ground on one side only, but greater care is necessary in its use as it is liable to slice the straight-edge. The knife should be used for all lines across the grain of the timber, while the point should invariably be used for lines that follow the grain of the timber. Having prepared the drawing tools it is now necessary to make length rods, on which all important centres should be marked, after which their faces should be varnished or, by repeated handling, it may be difficult to see the marks on the rod after it goes to the foundry, and length rods are invariably sent to the foundry. Too much care cannot be expended on the length rods. In very large castings it may be necessary to vary contraction allowances for length and height and width, and this is determined when making the length rods.

With regard to centre lines it is usually advisable to draw the main longitudinal centre line first. This, in the case of a narrow board, may be done by means of a final gauge from the edge, but more often a straight-edge is used, points an equal distance from the edge having previously been marked. The main cross centre should be drawn geometrically, after which other centre lines can be set down with trammels and straight-edge. We are describing this process because method is quite necessary in setting down work.

When all the important centre lines have been set down it is wise to check them from the drawing before starting the views. The elevation is usually the most important view. Although one view should be finished before another is started, it is as well when trammels are set to a certain dimension to transfer the dimension to the plan. Thus, in the case of a plain cylinder, a longitudinal section and a cross-section would be drawn, and while the trammels were set to the radius for the elevation the cross-section should also be drawn.

To make a neat and clean drawing, as few lines as possible should be lined in before a whole view has been finished, and no view should be coloured until the whole drawing is complete. All machined faces should be shown, and if camber has to be allowed the pattern should be drawn to the camber shape. In short, a workshop drawing should show the work as it will leave the pattern-shop. It is quite futile to set out work if additional allowances have to be made when constructing the pattern or core-boxes.

During the course of construction it is necessary to lay down additional lines showing timber joinings and divisions of cores. Extra lines of this kind should be as few as possible, because a dozen or more men may be working from one board and, if all are allowed to set out constructional lines, the value of the drawing will be considerably reduced. The one who has set out the work in the first place should be the one allowed to put additional lines down. Unless when it is necessary to develop a shape geometrically and when it can be much more easily done on the main board, constructional views should be set out on separate small boards.

When the various views have been set out and coloured, different colours being used for each view, the construction has to be decided upon. In regard to colouring the views, it is chiefly necessary to distinguish between the various sections, thus the metal of one may be coloured red, of another yellow, of another green, and so on. Boards should be varnished before the work is started. Some pattern-shops neither distinguish the separate views by colouring, nor varnish them, but it is slovenly practice not to do so.

It is impossible to start a squad of men on a large job until the construction has been thoroughly considered. Details may, of course, be left in many cases, but it should be remembered, that what may appear cursorily to be of minor importance, may have a very important bearing on the construction, and neglect of its consideration at an early stage may necessitate modifications later on. It is as well, therefore, not to overlook even the smallest details. Some parts of the job may offer several alternative methods, and it is not always an easy matter to decide which is best. Moulding has to be considered, and difficulties in machining the castings may have to be overcome by slight modifications of details. It is occasionally necessary to have a conference of pattern-maker, moulder, and machineman. In some shops it is customary for each man, as he makes a part, to fit it and secure it in position. Better results are obtained, however, when two men build the work and they are fed by the rest of the squad. If the job has been properly set out and care is taken by the men making the various parts, very little tool work should be required when erecting.

A custom which is not as common now as formerly, was for the man in charge of a large job to break out all the timber. So expert can one become with practice that the smallest bosses and prints can be rough cut. There is very little waste when this practice is adopted, because ends and corners that would be thrown on the scrap-heap are used for small ribs, bosses, and prints.

When erecting a large pattern, the main body must be set down to centre lines on the floor, or on the skids, first. Small blocks or buttons will keep it firmly in position. It is a good practice to set down centre lines and also lines representing flange faces, etc., on the floor, so that when branches or other parts are fitted a plumb bob may be used for testing. The plumb bob and the level are invaluable when building large patterns, and much more reliable than the try square or set square used from adjacent faces or from the floor.

When constructing large patterns, parts that the moulder will have to loosen off should always be screwed in such a manner that the moulder will have no difficulty in releasing them when the pattern is in the sand, and it is imperative that all screws should be greased so that they will unscrew readily.

(Conclusion)

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XXIX

Drawing Office Routine. The art of machine construction and drawing lends itself to the expression of individuality, but, as in many other arts, commercial considerations set limits within which the designer must work. One set of restrictions surrounding the draughtsman in practice may be placed under the title *drawing office routine*, and we may profitably give some attention to this aspect of the art.

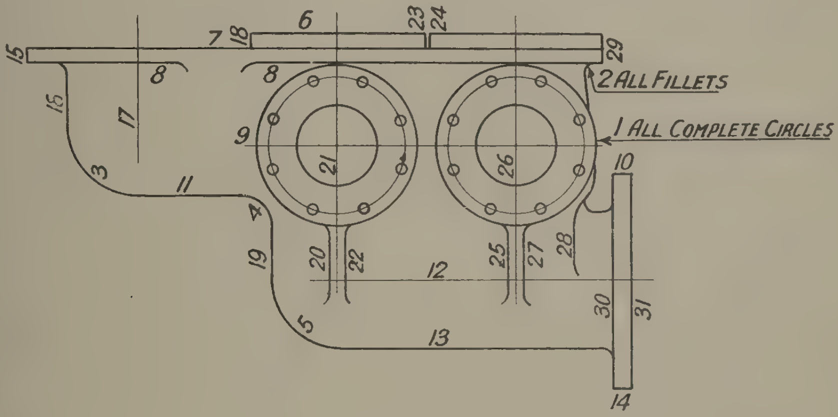


FIG. 2

Drawing office routine is not stereotyped, but a brief résumé of some of the more common features may serve as a rough guidance to the young designer who has not graduated through a large engineering drawing office.

Registration. We have, in a previous lesson, referred to the method of noting machine drawings by giving a contract number together with a drawing number embodying the contract number. Where drawings of a particular title are common to many contracts reference is assisted by allocating a corresponding number to each title. Such common notation is of great assistance to the draughtsman when searching for suitable guidance drawings.

Copying Drawings. Copies of drawings usually consist of photographic prints taken from a tracing of the drawing. The printing machine used for this purpose is specially designed for copying drawings of standard overall dimensions. The process of copying is simple and comparatively quick, the required time of exposure being constant for the power of light used and the type of print required.

The quality and durability of the print is chosen to suit the service required of the drawing. When the print is required for a moderate amount of perusal or reference, as in the case of prints submitted to owners for approval, a plain paper print may be used. When the drawing will be in frequent demand and have to sustain a lot of handling, as in the case of prints issued to the workshops, a more substantial print is necessary, and usually consists of a copy printed direct upon linen, or a paper print mounted upon cloth.

If, as often happens, the drawing is made upon a semi-transparent paper, called tracing paper, the copy may be printed direct from the pencil drawing. The pencil lines, however, do not produce the clearest copy and, generally, the only legible print to be obtained direct from the drawing is a blue print consisting of a white line on a blue ground, as shown in Fig. 1.

For more distinct copying it is necessary to make an ink tracing of the drawing, and if the tracing is to provide prints at a future date the tracing should be made upon transparent cloth.

Ink Tracing. In most large engineering establishments the ink tracing is not made by draughtsmen, but by tracers who, specializing in this section of the draughtsman's art, acquire a neatness and facility not usually obtained by the draughtsman. Apart from the question of speed there is no reason why a draughtsman should not produce an ink tracing of a drawing equal to the best product of the tracing office. Having a trained comprehension of the drawing he ought to proceed with the tracing without hesitation.

The elementary rules to be observed when tracing are self-evident. For example, it is much easier to join two arcs with a straight line than to join two straight lines with an arc. Further, ink can take a tantalizingly long time to dry, and one has, of course, to avoid moving the set square over traced lines before they are quite dry. An example is shown in Fig. 2, the numbers indicating the order in which the lines of this drawing may be traced for both neatness and economy of time.

Coloured Drawings. For direct indication of the materials used in the construction of a machine part the coloured drawing, as shown in Fig. 3, is most effective. The colouring not only shows at a glance the materials to be used in the construction, but also provides a drawing which is pleasing in appearance and helpful in conception. The print commonly used for the coloured drawing is the *black and white*, having black lines on a white ground.

TABLE I

Material	Colour Used
Brass	Gamboge
Bronze	Gamboge and burnt sienna
Copper	Burnt sienna
Cast iron	Payne's grey
Lead	Indigo
Steel	Crimson lake and Prussian blue
Cast steel	Crimson lake and Prussian blue with greater proportion of blue
Wrought iron	Prussian blue
Brickwork	Crimson lake and burnt sienna
Cement	Burnt umber
Concrete	Mottled grey and umber
Glass	Green
Woodwork	Pale tint of burnt sienna
Water	Prussian blue

Colours having a fairly close resemblance to the natural colours of the materials are used, a list of corresponding colours in common use being given in Table I.

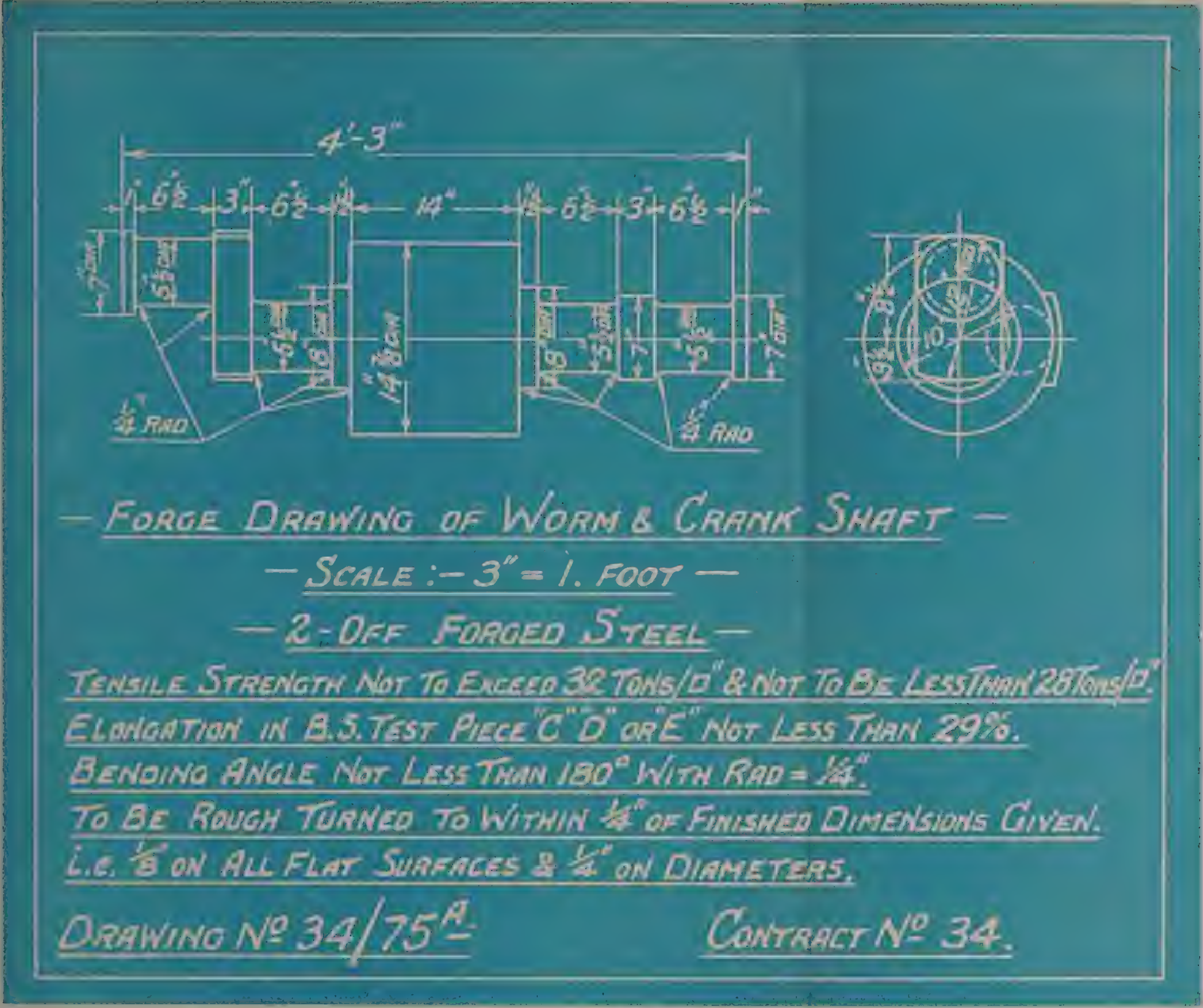
The definite line on a drawing makes neat colouring a simple operation, and this part of the draughtsman's art requires little practice to produce a tolerable effect. When the drawing is required for tendering purposes, appearance is more important, and the draughtsman would be advised to observe the elementary rules of artistic effect.

Drawing Office Copy. The most common and most important print of a drawing is the D.O.C. This print should bear a true record of the drawing as issued to the shops, and if additions or alterations are made to the shop copies a record of the change must immediately be embodied on the D.O.C. Sometimes the D.O.C. contains a record of the date of issue of the drawing to the various shops, but usually the record kept in the drawing register is considered sufficient.

Shop Copies. A print of the drawing is issued to each of the workshops through which the parts embodied on the drawing will pass in the process of manufacture. The drawing thus becomes an agent in the progress of each part through the shops, each drawing giving the reader an idea regarding its source when entering and its destination when leaving the shop.

When, as it often happens, a shop deals with only one of the smaller items on a drawing, a separate sketch of the part, and not a complete drawing, is issued to this shop. Such sketches can most conveniently be made from a copying-ink sketch, and need not be drawn correctly to scale. Further copies of the sketch are issued also to the shops having the drawing, in order to inform these shops of the progress of the part. A typical sketch of the worm and crankshaft for the steam steering gear is shown in Fig. 5.

Sub-contractors. When a portion of the work on a drawing is made by an outside firm it is advisable to provide a sketch of the part, as required from the makers, and not the complete finished working drawing. For example, if the worm and crankshaft shown in Fig. 4 is to be ordered in the rough from an outside firm, Fig. 4 may be replaced with the rough machine drawing shown in Fig. 1.



— FORGE DRAWING OF WORM & CRANK SHAFT —

— SCALE :– 3" = 1. FOOT —

— 2 - OFF FORGED STEEL —

TENSILE STRENGTH NOT TO EXCEED 32 TONS/□" & NOT TO BE LESS THAN 28 TONS/□".

ELONGATION IN B.S. TEST PIECE "C" "D" OR "E" NOT LESS THAN 29%.

BENDING ANGLE NOT LESS THAN 180° WITH RAD = ¼".

TO BE ROUGH TURNED TO WITHIN ¼" OF FINISHED DIMENSIONS GIVEN.

I.E. ⅛" ON ALL FLAT SURFACES & ¼" ON DIAMETERS.

DRAWING Nº 34/75^{A} CONTRACT Nº 34.

FIG. 1. FACSIMILE OF BLUEPRINT

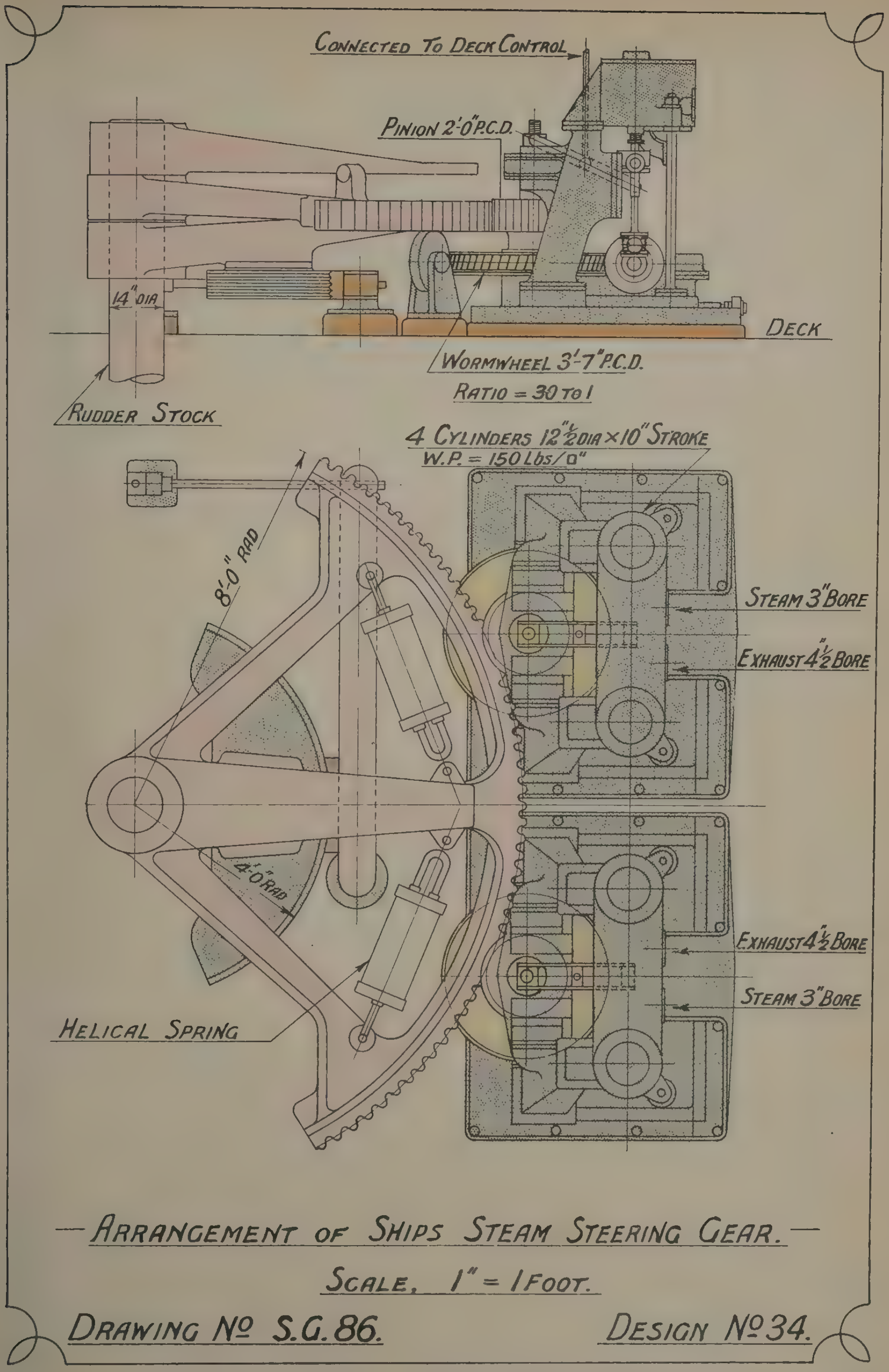
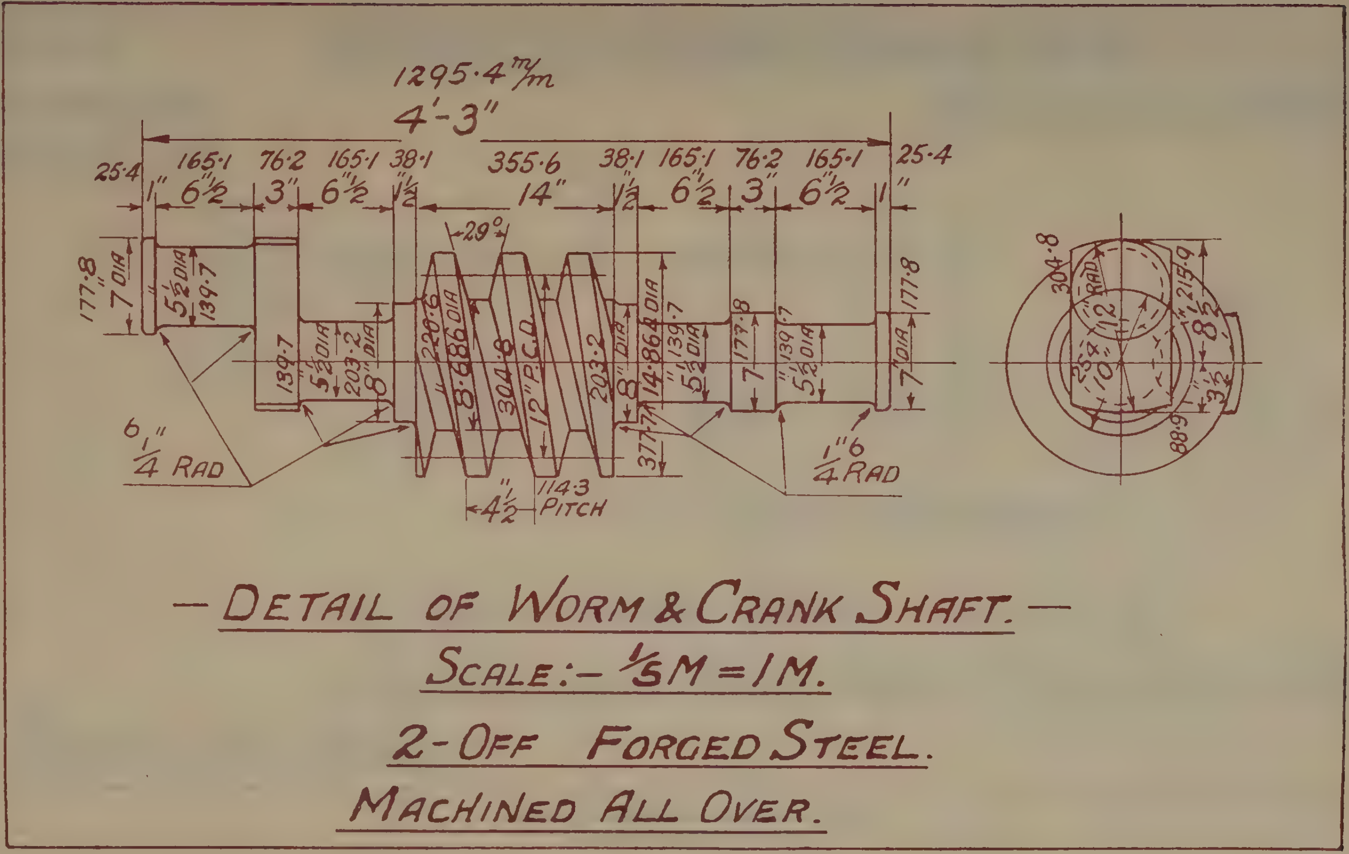


Fig. 3. Facsimile Coloured Drawing of Steam Steering Engine



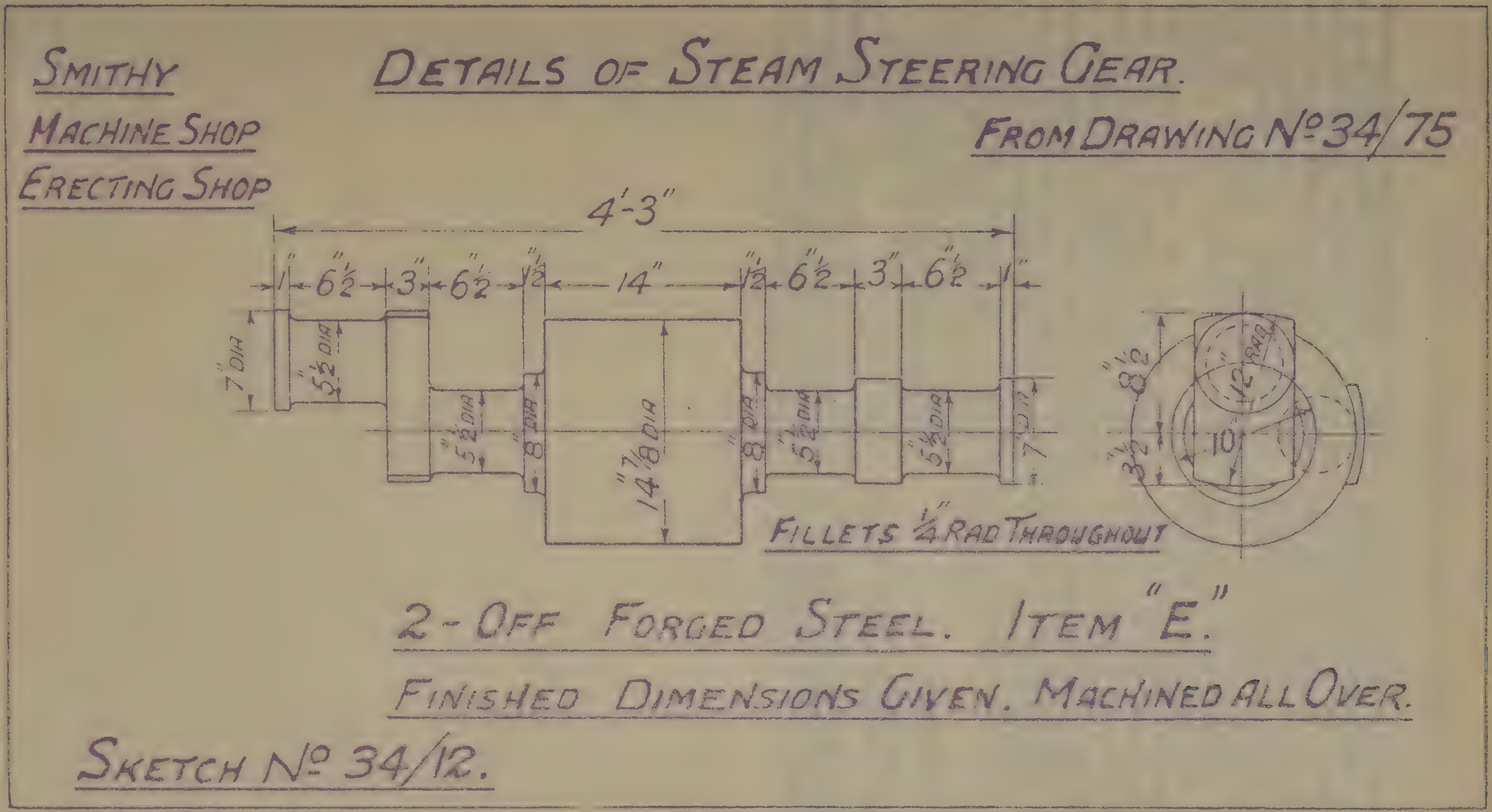
— DETAIL OF WORM & CRANK SHAFT. —

SCALE:– 1/5 M = 1 M.

2-OFF FORGED STEEL.

MACHINED ALL OVER.

FIG. 4. FACSIMILE OZALID OR CORALIN PRINT



SMITHY

MACHINE SHOP

ERECTING SHOP

DETAILS OF STEAM STEERING GEAR.

FROM DRAWING Nº 34/75

FILLETS ¼" RAD THROUGHOUT

2 - OFF FORGED STEEL. ITEM "E."

FINISHED DIMENSIONS GIVEN. MACHINED ALL OVER.

SKETCH Nº 34/12.

FIG. 5. FACSIMILE COPYING-INK SKETCH
(Made by the gelatine pad process)

Drawing for Foreign Manufacture. Consulting work often demands drawings for use in foreign workshops, and we may consider the means taken to adapt our practice to suit the foreign workshop. It is usually found desirable to design as for British manufacture, giving the dimensions in British units and adding the metric equivalent above each dimension.

This method allows the designer to use British reference drawings, and to think in terms of the more familiar units of length. The example shown in Fig. 4 is an ozalid print taken by a process of Messrs. Agfa, Ltd., which leaves the scale of the drawing unaltered. This type of print is very useful where prints, being sent abroad, have to be traced to produce copies adapted to foreign workshop practice.

Messrs. B. J. Hall & Co. also reproduce drawings by a similar process known as Coralin. They do not, however, claim that this gives copies true to scale.

Conclusion. Although engineering practice, having made such rapid strides, appears to cover the whole field of novelty in design, the student need not be discouraged into thinking the art has developed into mere copying. The field is actually wider than ever, but the task of finding a new seam may be increasingly difficult and demand an alert perseverance. Also, one may be led to think that the outlook of most employers is hidebound by a conservative prejudice for what has been tried by experience. Such attitude is apt to discourage until one reflects that an employer's whim is usually based upon his particular experience. Further, one may often learn from what has the appearance of mere prejudice.

The fact remains that engineering design has a great future, and the difficulty of attainment ought to induce the ambitious student to master the art and, at some future time, make the impress of his individuality upon what is a very fine useful art.

EXERCISE

Using Fig. 3 as a design basis plan, and referring to previous lessons, complete the dimensioning of the plan.

Next, make detail drawings of as many of the parts as the student's progress and initiative will permit. Finally make ink tracings from some of the more simple drawings.

(CONCLUSION)

GAS ENGINES AND PRODUCERS

By Arnold Rimmer, B.Eng.

LESSON XIII

POWER GAS PLANTS

Suction Plants

"National" Anthracite Plant. This plant, designed to burn anthracite or coke fuel, is illustrated in Fig. 70, a section of the producer and scrubber being shown in Fig. 71.

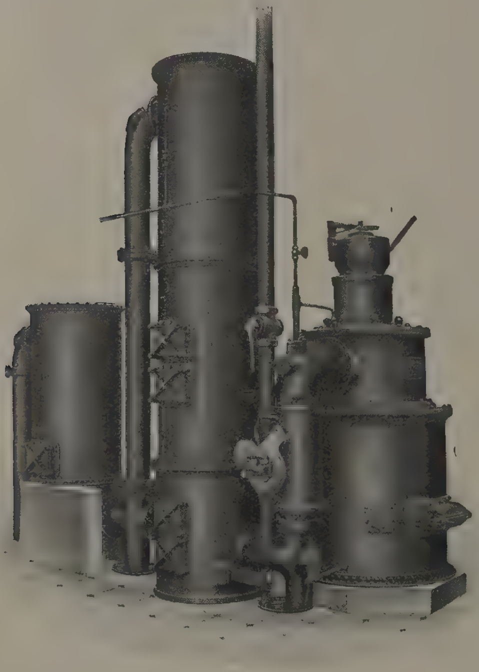


Fig. 70. National Suction Gas Plant (Anthracite Fuel)

The generator consists of a plain cylindrical casing lined with firebrick. The fuel is fed in through a hopper of large capacity, arranged with double valves which prevent leakage of gas or air. Water is fed from a trough at the top of the vaporizer and evenly distributed over the outside of the latter, where it is evaporated. Air enters the space between the vaporizer and outer casing, mixes with the steam, and flows through a passage to the underside of the grate. Additional air is admitted at this point, and the combined steam and air then pass through the fuel bed.

The gas formed then passes to the coke scrubber, which is a steel cylinder with a cast-iron base (to reduce the effect of corrosion). Water is distributed through a sprinkler at the top and percolates through the coke, thus washing the impurities from the ascending gas. A small "purifying scrubber" may also be fitted if the fuel to be used is of inferior quality.

A hand-driven fan is provided for blowing in air when starting up the plant, the gas generated meanwhile escaping through the waste pipe. The latter is closed when the gas is of sufficient richness to be passed to the engine.

The grate is fitted with conical type fire-bars, designed to facilitate the removal of clinker, and poking holes are arranged in the side and top of the generator.

In larger plants the vaporizer is of the flash type and is fixed outside the generator, the telescopic outer casing allowing of easy access to the inner tubes (Fig. 72). A dry scrubber, containing coke, wood, wool, and sawdust is also provided, and this ensures the gas being absolutely dry and clean before passing to the engine.

Crossley Bituminous Fuel Plant. The gas plant manufactured by Messrs. Crossley Bros., Ltd., for the utilization of bituminous fuel is illustrated in Fig. 73, a section of the same being shown in Fig. 74.

It is of the closed-hearth type, with a conical grate consisting of a number of overlapping plates. These allow of an even distribution to the fire of air and steam, while preventing ashes choking the supply of the same.

The feeding hopper is of the rotary valve type, and this is combined with an "air-lock" hopper if it is desired to burn waste wood with bituminous coal.

The vaporizer is situated at the base of the generator, thus taking full advantage of the heat from the fuel bed and reducing radiation losses. In addition, the top of the generator is left clear of all apparatus which might tend to foul with dust and tarry products of combustion. Cleaning doors in the outer casing allow of access to the vaporizer for the removal of scale, etc., and poking holes, fitted with taper plugs, are provided in the top of the generator. The easy removal of ashes, without the entrance of air, is rendered possible by the water seal or "lute" surrounding the generator. Hence, the plant is capable of continuous working over long periods.

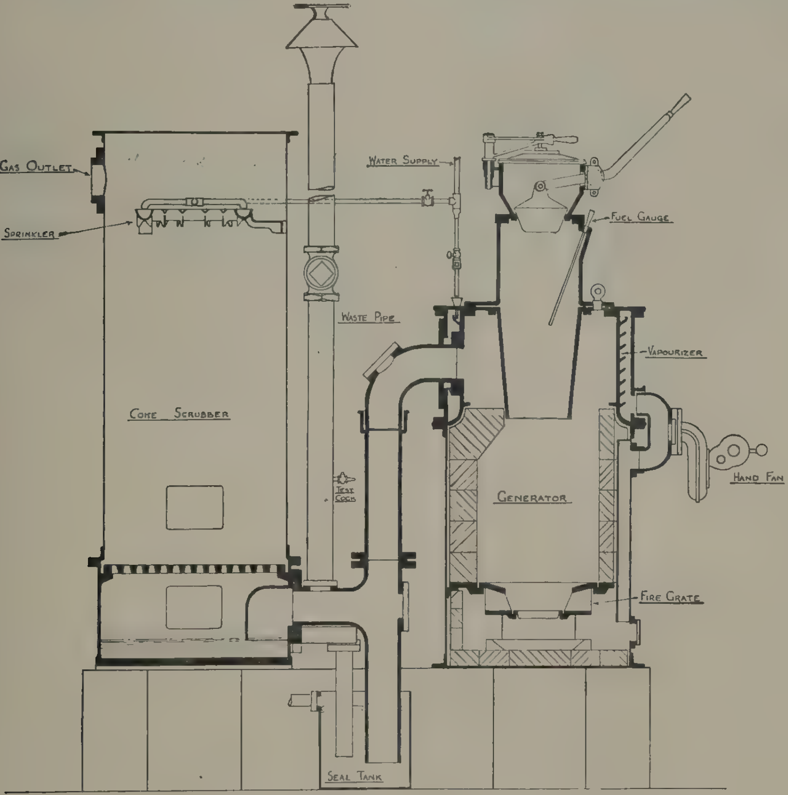


Fig. 71. National Suction Gas Plant (Anthracite Fuel)

In the pipe leading from the generator is fitted a water spray which effectually removes any heavy dust in the gas and deposits it in the seal box at the bottom of the pipe. The gas then passes through a wet scrubber to the centrifugal "tar extractor." The latter consists of a double bladed fan impeller revolving at high speed in a cast-iron casing. A small quantity of water is fed in and forms, with the tar "beaten out" of the gas, an emulsion, which is run off through the seal pipe in the base. The gas is then passed through a large cylindrical filter (containing layers of wood, wool, or similar loose material supported on wooden grids), after which it is in a thoroughly clean, cool, and dry condition.

Waste Fuel Plants. Reference has already been made to recent developments in the utilization of waste fuel. The gas plants designed for dealing with the latter are similar, in general construction, to those already described. The producer is usually of the closed-hearth type and, as a rule, no vaporizer is required, as the fuel itself generally contains sufficient moisture to give the necessary steam.

The waste fuel gas plant as manufactured by the National Gas Engine Co., Ltd., is illustrated in Fig. 75, and a section of that made by Messrs. Crossley Bros., Ltd., is shown in Fig. 76.

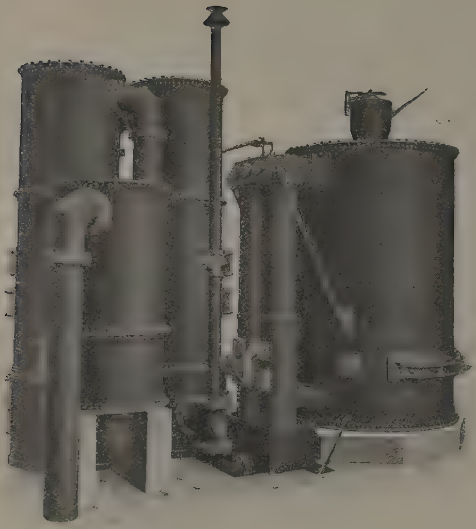


Fig. 72. National Suction Gas Plant (Anthracite Fuel)

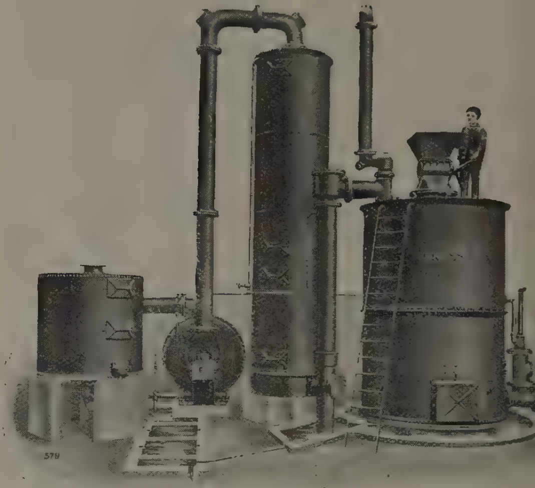


Fig. 73. Crossley Suction Gas Plant (Bituminous Fuel)

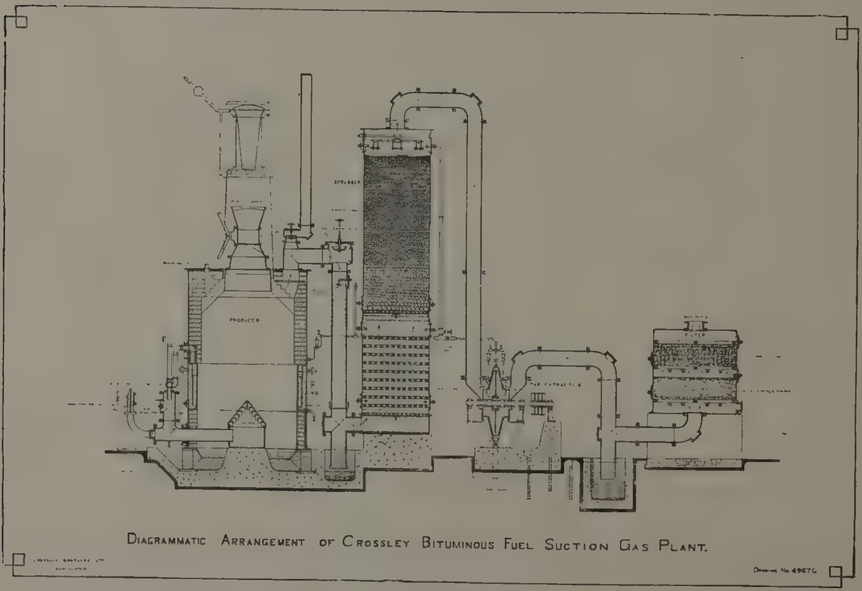

SCRUBBER
PRODUCER
TAR EXTRACTOR
FILTER
Diagrammatic Arrangement of Crossley Bituminous Fuel Suction Gas Plant.
Drawing No. 49676

Fig. 74

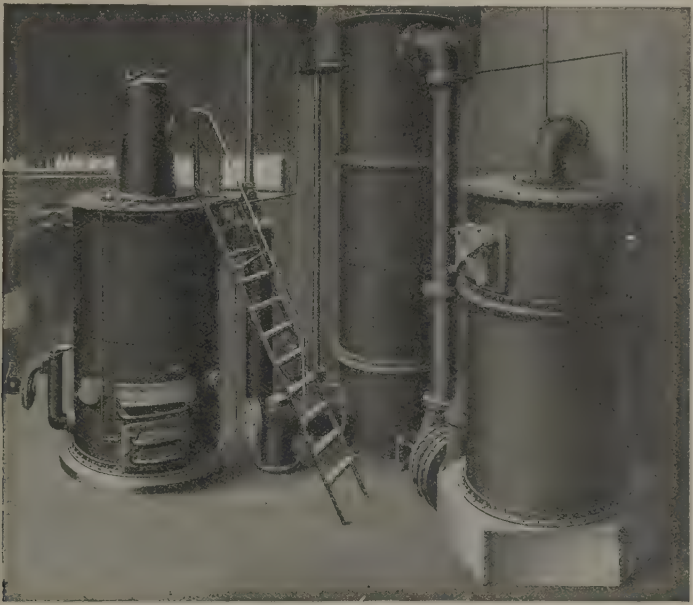
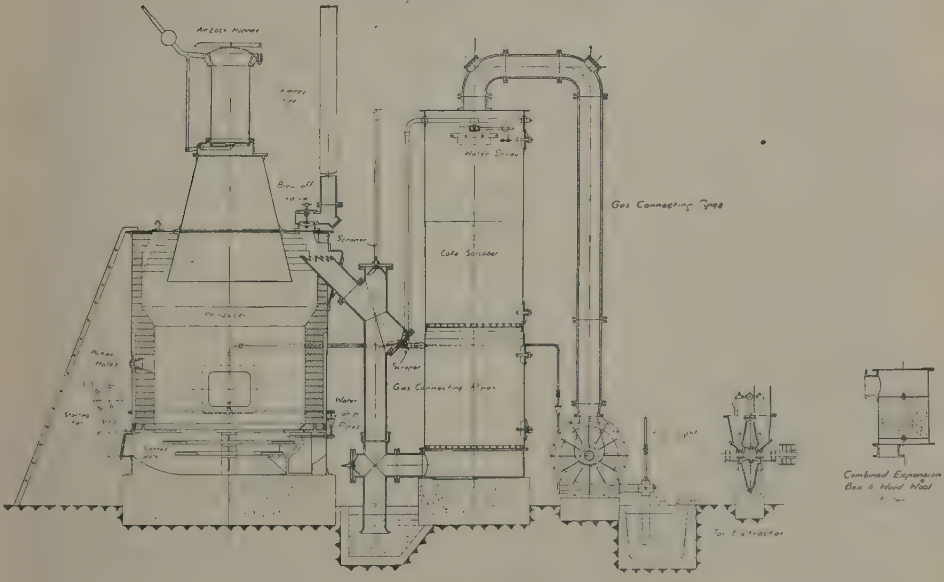


FIG. 75. NATIONAL SUCTION GAS PLANT (WASTE FUEL)



DIAGRAMMATIC ARRANGEMENT OF "CROSSLEY WOOD FUEL SUCTION GAS PLANT

FIG. 76

A feature of the latter plant is the stepped grate, consisting of a number of circular flat-plates, upon which the fuel is supported in sections. (See also Fig. 77.)

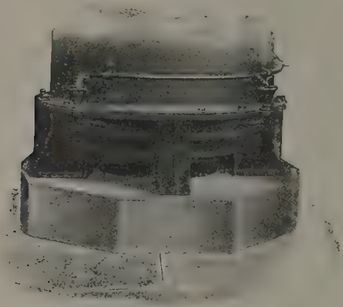


FIG. 77. STEPPED GRATE (CROSSLEY)

Since these plates lie outside the angle of repose of the fuel, none of the latter will of itself fall from the grate, while the fuel rests on its own bed of ashes beneath the bottom plate. A large area is provided for the passage of air, etc., through the fire, and the condition of the same can be readily ascertained and ashes, etc., removed without interrupting the process of gas making.

PRESSURE PLANTS

Although the suction gas plant has now an extensive application, there still remain cases in which it is desirable that the gas shall be supplied to the engine at a pressure in excess of that of the atmosphere. The "Dowson" and the "Mond" plants are two well-known examples of this type.

Dowson Gas Plant. In the original Dowson plant, steam from a separate boiler was blown through a special jet into the incandescent fuel, air being induced at the same time. While a gas of slightly higher calorific value was obtained, the capital cost was considerably greater, and the Dowson and Mason Gas Plant Co., Ltd., Levenshulme, Manchester, do not find much demand for this type of plant at the present time.

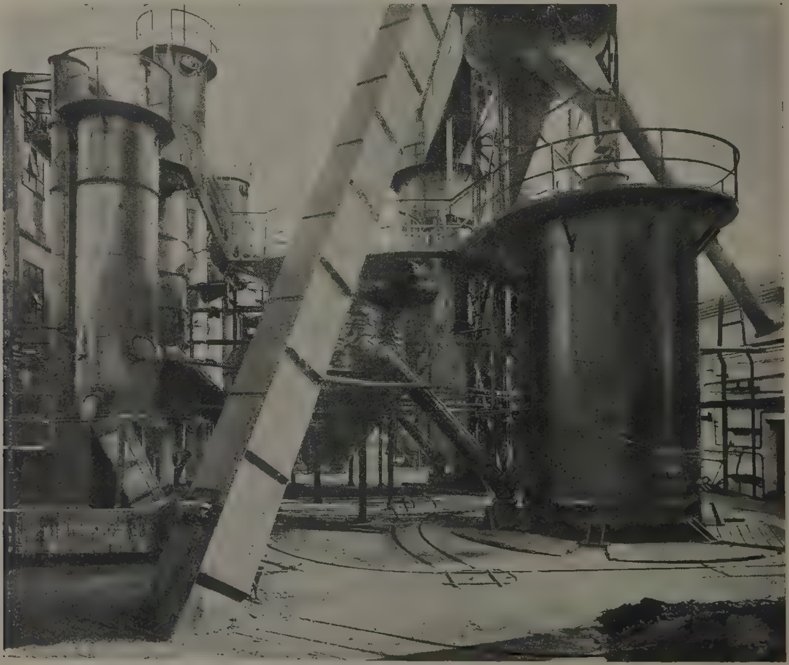


FIG. 80. "MOND" RECOVERY GAS PLANT
(Power Gas Corporation)

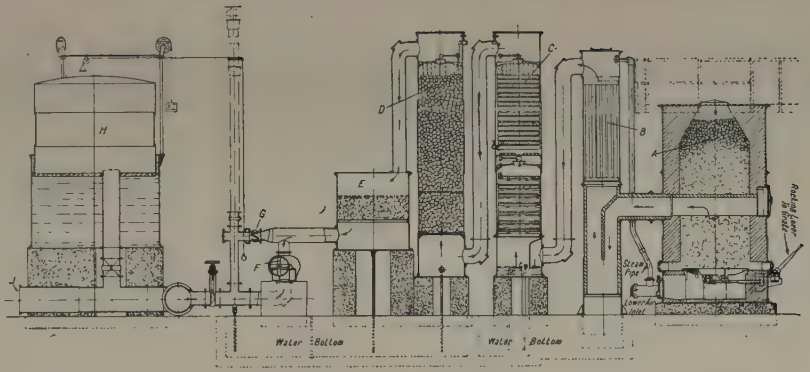


FIG. 78. DOWSON UP AND DOWN DRAUGHT GAS PLANT (BITUMINOUS FUEL)

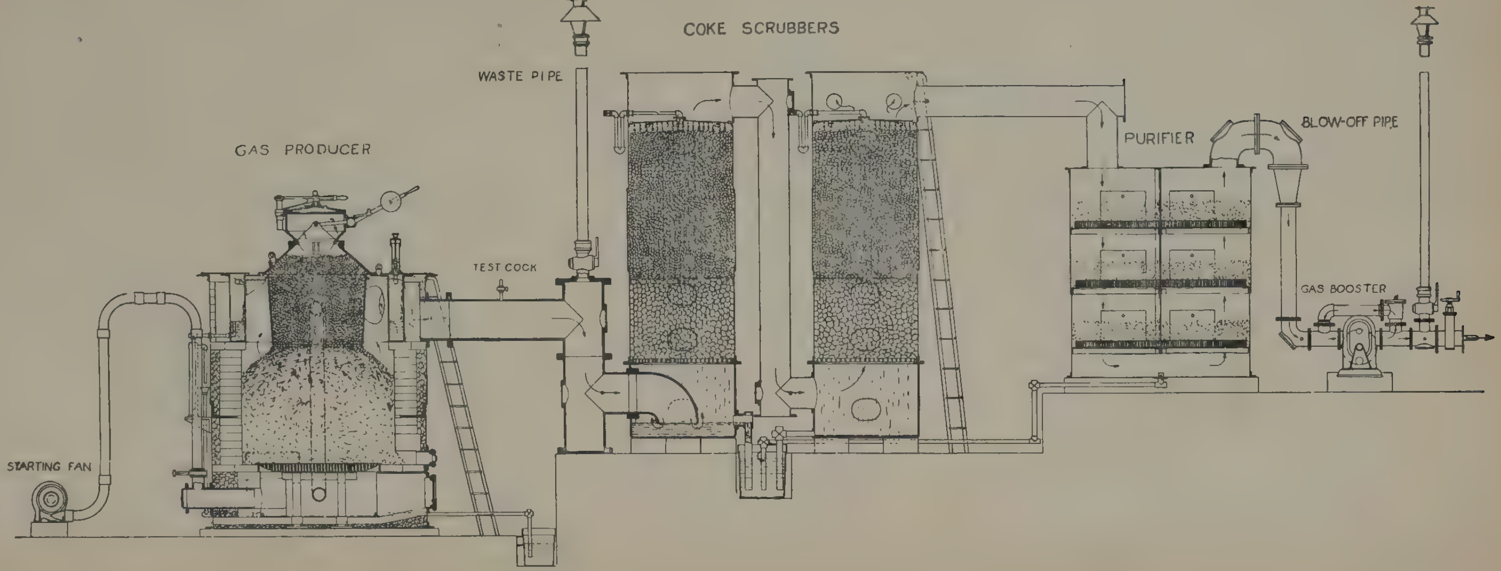


FIG. 79. DOWSON OPEN HEARTH GAS PLANT AND BOOSTER

By the kindness of the above firm, a section of one of their plants, also associated with the name of Mr. Dowson, is shown in Fig. 78.

It is designed to burn bituminous fuel, and is of the closed-hearth type, the fire-bars being connected to a rocking lever by means of which undue accumulation of ashes may be prevented. The generator is open at the top and, at about mid-height, is provided with an inverted U-channel of refractory material. Gases formed from the upper layers of fuel, as it is fed in, pass downward, but have their tarry vapours, etc., destroyed before mixing with the gas arising from the lower portions of the fuel bed. The gas then passes via the external vaporizer *B* to the scrubbers *C*, *D*, and *E*, and then to the blower *F* (which draws the gas through the system). A gas-holder *H* is provided and this is connected to a regulating valve *G*, by means of which gas may be by-passed from the delivery to the suction side of the blower when the supply of gas exceeds the demand.

In Fig. 79 is shown an open-hearth plant, made by the same firm, to use anthracite as fuel.

It is combined with a booster by means of which the gas may be delivered at a pressure up to 80 in. of water. As will be seen, a more elaborate cleaning plant, including an oxide purifier, is provided in this case.

Mond Gas Plants. The sole manufacture of these plants, under the Mond and Lymn patents, is carried out by the Power Gas Corporation, Ltd., Parkfield Works, Stockton-on-Tees, by whose courtesy they are here described.

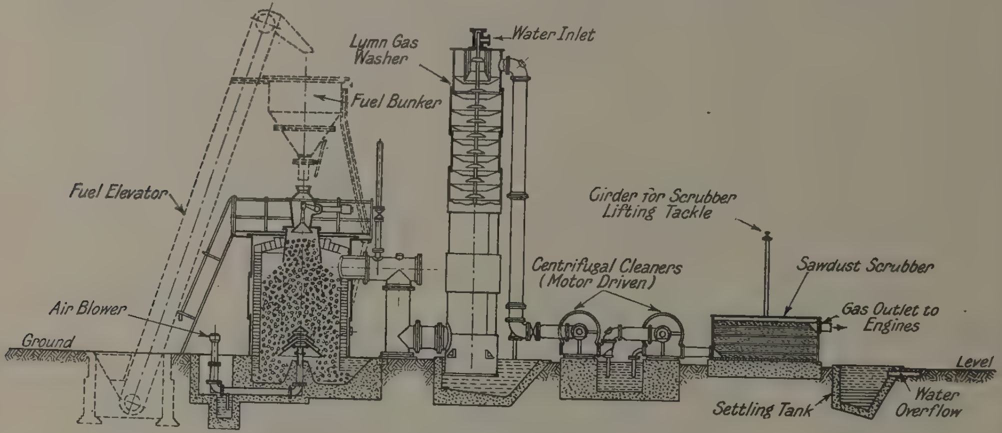


FIG. 81. MOND NON-RECOVERY POWER GAS PLANT
(Power Gas Corporation)

The fuel used may be ordinary bituminous slack or coal, colliery refuse, etc., and a complete plant may include arrangements for the recovery of sulphate of ammonia and other by-products (Fig. 80).

In Fig. 81 is shown a diagrammatic section of a non-recovery plant, while an illustration of the producer and accessories is shown in Fig. 82, and that of the gas washing and cleaning plant in Fig. 83.

The generator is fixed on a concrete foundation and provided with a water lute. The grate, which may be of the mechanical type as shown in Fig. 84, is of conical form, built up of cast-iron sections and provided with air slots, the blast pipe passing through the foundation from a blower of the steam jet type.

The feeding hopper has a sliding cover and balanced cone valve, and below this is a fuel container (provided with suitable poke holes. In this the gases from a fresh charge are distilled, and the tarry products have then to pass through a hotter zone, in which they are converted into fixed gases. Fuel-handling plant, as indicated, is frequently included in the producer equipment.

The Lymn gas washer and reaction tower is of patented design. It contains no moving parts, and requires a minimum of attention and upkeep. By means of discs fixed to a central shaft and inverted truncated cones attached to the outer casing, the momentum of the upward flowing gases is utilized to atomize water falling

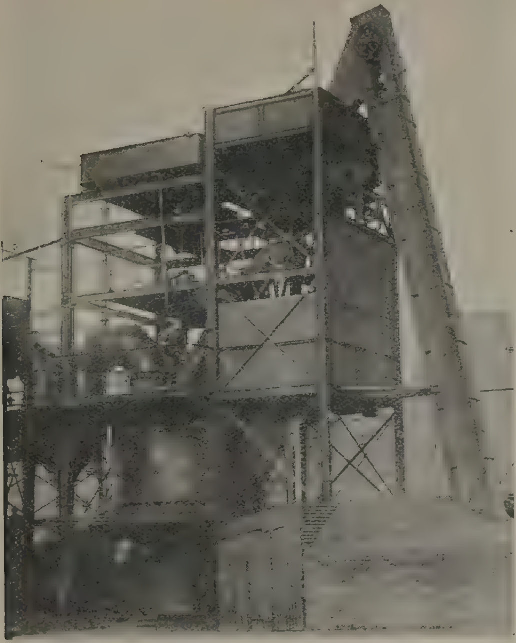


Fig. 82. Gas Producer with Mechanical Grate, Fuel Feed, and Stirrer
(Power Gas Corporation)

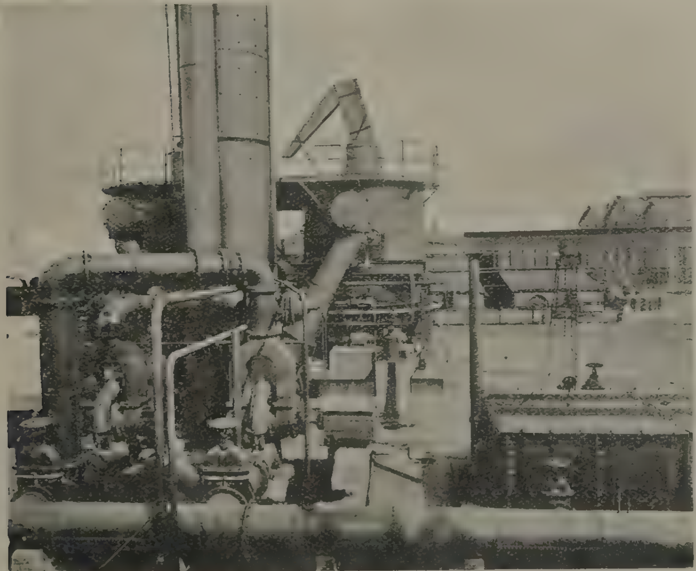


Fig. 83. Gas Washing and Cleaning Plant
(Power Gas Corporation)

in thin sheets, resulting in an intimate contact between the two fluids.

The two centrifugal cleaners are normally arranged to work in series (though by-passes, for cleaning purposes may be arranged), and each is provided with a tar-drain water sealed in a lute.

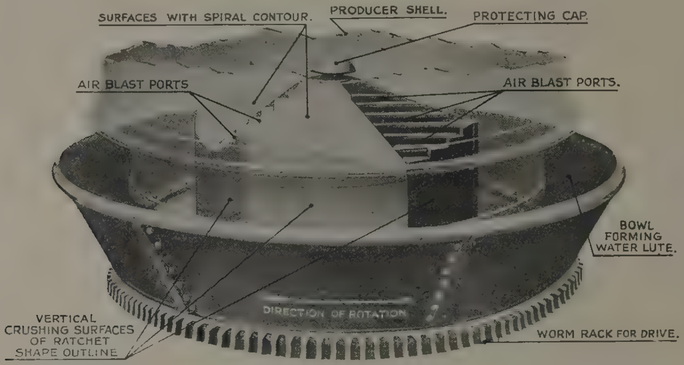


Fig. 84. Mechanical Grate
(Power Gas Corporation)

The scrubber is filled with shavings and sawdust (arranged on wooden grids), which only require renewal a few times per year. The spent material may be burnt under the steam boiler.

The drains from the various parts of the plant run to a settling tank from which the water escapes, and the tar is collected and utilized in various ways depending on local conditions.

The gas pressure may be kept very constant by fitting a small gas-holder, the bell of which, as it approaches its top position, actuates a relief valve on the blast main, thus reducing the pressure in the same and automatically controlling the production of gas according to the load on the engine.

(Conclusion)

STEAM BOILERS AND THEIR ACCESSORIES

By Percy E. Rycroft, M.B.E., M.I.E.E.

LESSON XII

DRAUGHT

THERE is nothing more important in connection with steam raising plant than the question of providing a suitable and continuous supply of air—or more precisely, oxygen—to the boiler furnace in order that efficient combustion of the fuel may be assured.

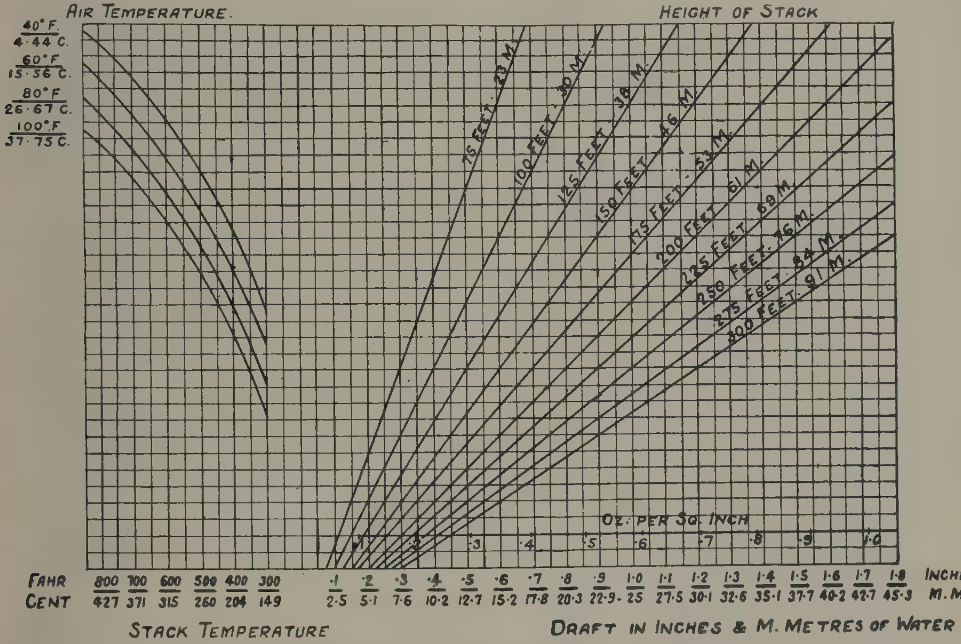


FIG. 77. CURVE SHOWING THE THEORETICAL DRAUGHT PRODUCED BY CHIMNEY STACKS OF DIFFERENT HEIGHTS WITH DIFFERENT TEMPERATURES OF FLUE GASES AND OF ATMOSPHERE

An ordinary chimney operating under what is known as " natural draught " conditions performs this function by reason of the fact that heated air occupies a greater space for a given weight than cold air.

If we visualize the column of air in a chimney and a column of air of similar dimensions outside, as the two legs of a U tube in which the boiler furnace forms the connecting portion, it will be seen that the inside column of warm air always weighs less than the outer column of cold air, and a permanent out of balance results. By this means air is continually forced by atmospheric pressure through the furnace, and a definite and measurable degree of draught is obtained, varying with the height of the chimney and the temperature of the gases entering the chimney.

Although, as has been said, it is really the pressure of the atmosphere which forces air through a furnace into the partial vacuum formed by the expansion of the heated air, it is more usual to talk of a *suction* draught at the boiler outlet, and this is conveniently measured and referred to in terms of inches of water.

As an example, " one inch of draught " is an air pressure or suction of sufficient intensity, when applied to one side of a U tube partially filled with water, to create a difference in water level between the two sides of one inch.

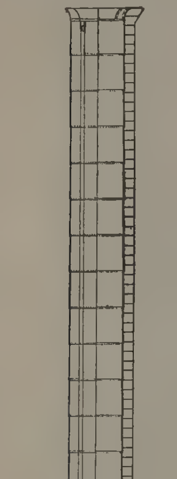


Fig. 78. Self supporting Steel Chimney, showing Cone at Base

In view of what has been said it will be evident that the higher the chimney the greater will be the difference in weight of the inside and outside columns of air and, consequently, the greater the intensity of the draught.

Similarly, it will be obvious that the hotter the temperature at which the flue gases are discharged to the chimney the less will be their density, and the greater their velocity, so that this consideration also has a material effect upon the draught obtained by "natural" means.

In the accompanying table the theoretical draught available from chimneys of varying heights, under given conditions of atmospheric and flue gas temperatures, can be readily calculated. As an example, if we desire to ascertain the draught of a chimney 150 ft. high with a flue gas temperature of 400° F., while that of the outside air is 60° F., we follow the 400° F. line vertically until it cuts the 60° F. air temperature curve. By carrying this point across horizontally until it meets the 150 ft. chimney line, and from here dropping vertically to the base line, we can read off the theoretical draught at the chimney base, viz. ·825 in.

After selecting a few such examples it will soon be seen that for a draught of anything over an inch and a half the height of chimney required, if natural draught alone is to be relied upon, becomes a matter for serious consideration, not only on account of mechanical difficulties of construction, but also for the cost involved. Moreover, although brick chimneys have been constructed up to a height of 400 ft., and many thousands between 150-250 ft. are still in use, the draught requirements of large modern steam raising plants can no longer be always satisfactorily met by "natural" means alone.

The addition of superheaters, economizers, air heaters, dust catchers, and other modern accessories to the boiler proper all offer additional resistance to the passage of the furnace gases through the unit, and it is now not at all uncommon for from 2½ in. to 3 in. of water gauge or even more to be allowed for at the chimney base to overcome this resistance. In addition the necessity for extracting the maximum possible percentage of heat from the flue gases, before allowing them to escape to atmosphere, deprives the chimney of any assistance due to high temperature, as many modern plants now exhaust their gases at very little above 200° F.

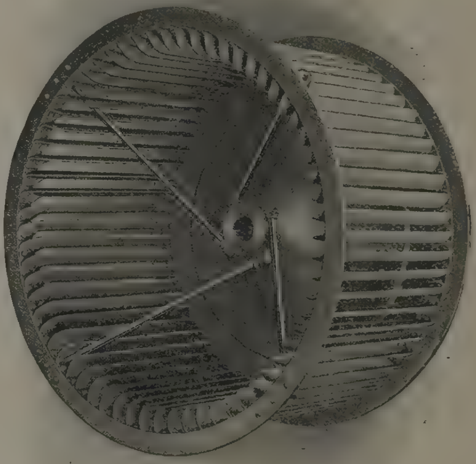


Fig. 79. Single-inlet Fan Impeller
(Davidson)

All these considerations would have involved chimneys of gigantic proportions and led naturally to the development of mechanical draught where, by means of electrically or steam operated fans inserted in the flue circuit, any desired draught can be readily provided, quite irrespective of the height of the chimney or of the temperature of the flue gases.

This being so, the relatively short, light, cheap, and quickly erected steel chimney has rapidly advanced in favour so that its use in all the larger plants is now universal, a separate chimney for each complete unit or pair of units being the common practice.

As to height it may be taken that if mechanical draught is to be relied upon entirely, as is the case with very large units, a distance of 60 ft. from the firing level to the top of the stack is sufficient. In moderate sized plants, however, where a certain proportion of the daily load can be adequately dealt with by means of natural draught, a height of from 100 ft. to 120 ft. should be provided as this will enable the fans to be shut down, and so reduce power expenditure during the lighter load periods.

Opinions vary a little as to the desirability or otherwise of lining steel chimneys with brickwork.

Generally speaking, there is no advantage in providing a lining except, perhaps, for a few feet from the base, where the additional weight adds to the stability of the structure. Care must, however, always be taken to prevent the possibility of moisture getting between the lining and the chimney plates, as more harm in the form of corrosion is likely to occur under this condition than if the chimney is entirely unprotected so far as the introduction of a lining is concerned. Steel chimneys are either parallel from base to top and stayed by means of guy ropes or are coned or gusseted at the base in order to be self-supporting.

Mechanical draught may be classified under three headings, (1) Induced, (2) Forced. (3) Balanced.

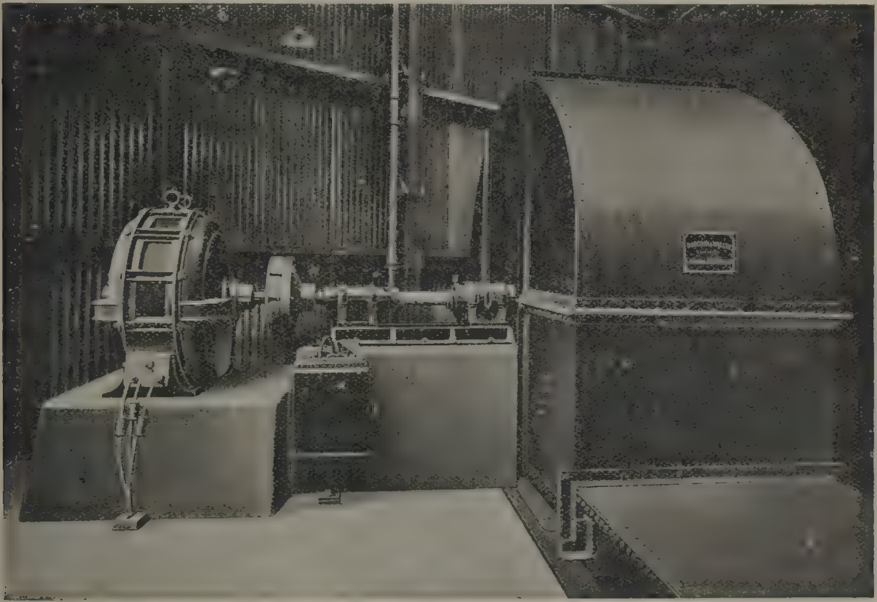


Fig. 80. Direct-coupled Motor-driven Induced Draught Fan
(Babcock & Wilcox, Limited)

Simple induced draught is the most common and can usually be added to existing installations with very little trouble. Forced draught generally involves the closing in and sealing of the ash-pits, or underside of the grate furnace, but may be provided by means of a steam blast, while balanced draught is probably the most efficient of all and is a combination of both induced and forced.

Induced draught fans are either electrically or engine driven, and are located either at or close to the base of the stack.

It is the best practice to arrange a by-pass flue, so that the flue gases can be exhausted direct into the chimney without passing through the fan, when it is not necessary for the fan to be employed.

It should be noted that owing to the temperature of the gases with which the fan deals, a considerable amount of heat is conducted along the fan and motor shaft, and the fan bearings are, therefore, usually water-cooled.

The most general arrangement is for an overhung fan to be employed, drawing in the flue gases at the centre and discharging them through an outlet in the casing provided at some point of the fan's periphery.

Wherever possible, induced draught fans should be provided with some means of speed regulation.

With a steam-engine or direct current motor this is a simple matter to arrange, but with alternating current motors the problem is rather more difficult. Fans can be belt or chain driven, but a direct couple to the prime mover is the best arrangement.

By the installation of an induced draught plant and the provision thereby of an increased suction draught through the furnace a higher consumption of fuel per square foot of grate area and, consequently, a higher evaporation can be obtained from a given plant. The degree to which this additional evaporation can be extended is, of course, limited by a number of factors and must not be unduly stressed, but the advantage of a flexible draught for dealing with poor quality fuel and fuels low in volatile matter will be apparent. It must be noted, however, that as the suction draught is increased so the liability of infiltration of air through furnace brickwork and flue leaks becomes intensified, and it is of particular importance to prevent such leakages if efficiency is to be maintained.

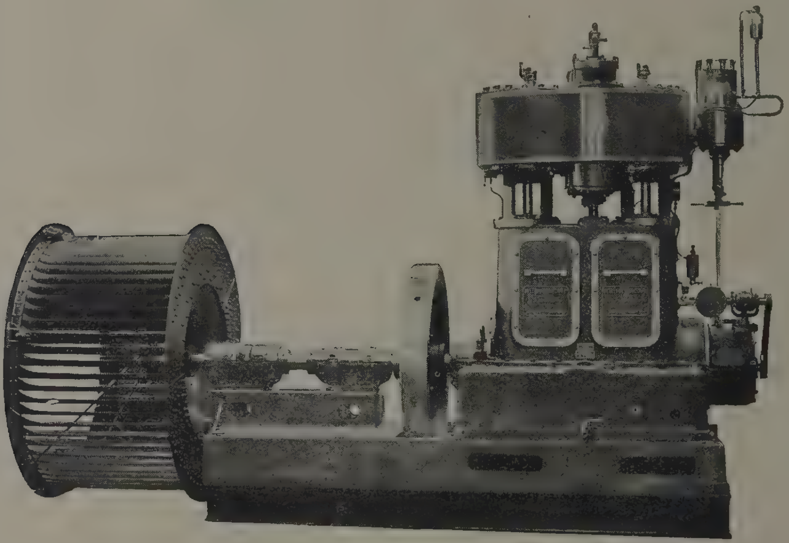


FIG. 81. "SIROCCO" FAN COUPLED TO STEAM-ENGINE

(Fan Casing Removed)

(Davidson)

The primary function for forced draught is to overcome the resistance of the fuel bed to the passage of air through the grate, and where heavy fires are employed this resistance may require a water gauge pressure of as much as 3½ in. to 4 in. to overcome it although from 1 in. to 2 in. is sufficient for most conditions.

The best method of applying forced draught is to enclose the ash-pit and utilize fans for creating the necessary air pressure. Steam jets are, however, often used to blow air directly between the grate bars, more particularly with hand-fired installations. Such a steam and air blast is quite effective in the burning of low grade or low volatile fuels, but has the demerits of being noisy and somewhat wasteful of steam.

A consumption of 5 per cent of the boiler output is a not uncommon figure when steam jets are used, and this can be easily exceeded as the cutting action of the steam speedily enlarges the nozzle orifice.

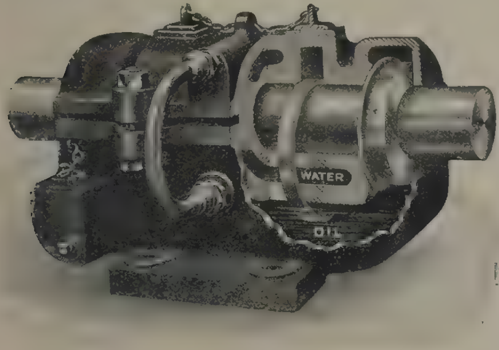


FIG. 82. WATER-COOLED FAN BEARING
(Davidson)

With the larger modern plants the use of balanced draught is being very largely adopted.

In this case the furnace ash-pits are enclosed or a series of air-tight compartments are arranged directly under the fire-grate, into which air is delivered under pressure by means of a forced draught fan.

Sufficient pressure is provided to overcome the resistance of the fuel bed, while the suction draught is so regulated that it will just pull the gases through the boiler, the draught in the actual furnace being practically balanced. In some of the latest designs of fire-grate, arrangements are provided whereby the under-grate pressure over various sections can be regulated independently, and this is a particularly valuable feature when varying grades of fuel have to be dealt with.

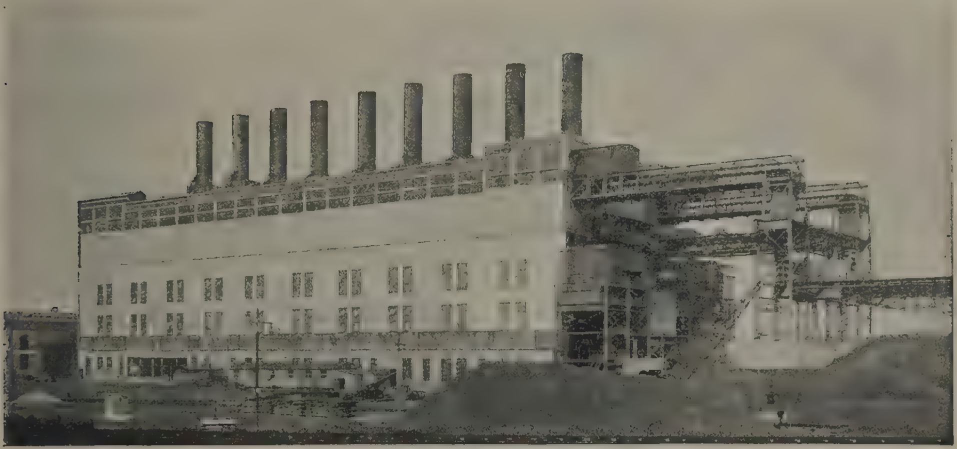


FIG. 83. TYPICAL MODERN SUPER ELECTRIC POWER HOUSE, SHOWING BATTERY OF STEEL CHIMNEYS, EACH BOILER UNIT HAVING A SEPARATE STACK
(Davidson)

(CONCLUSION)

REFRIGERATING MACHINERY

By G. W. Daniels, M.Eng., Wh.Ex., A.M.I.Mech.E.

LESSON V

COLD STORAGE

One of the principal duties of refrigerating machinery on land is the cooling of the cold stores which contain our food supplies.

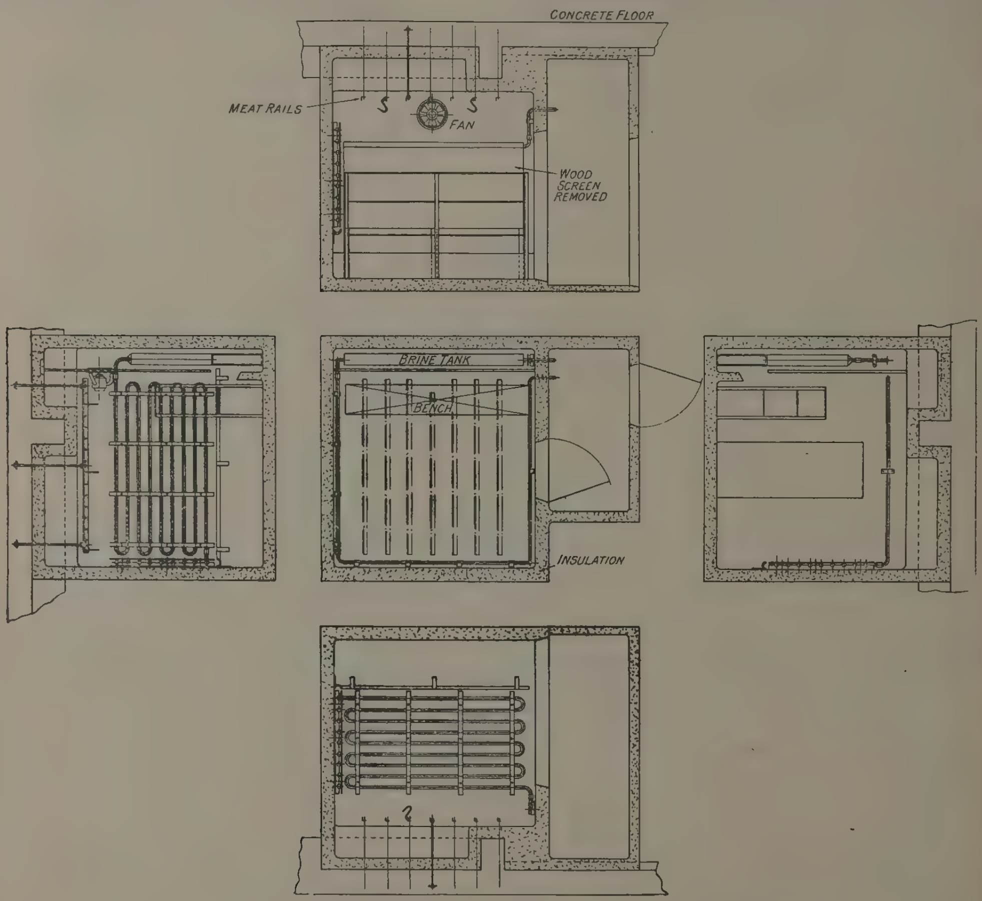
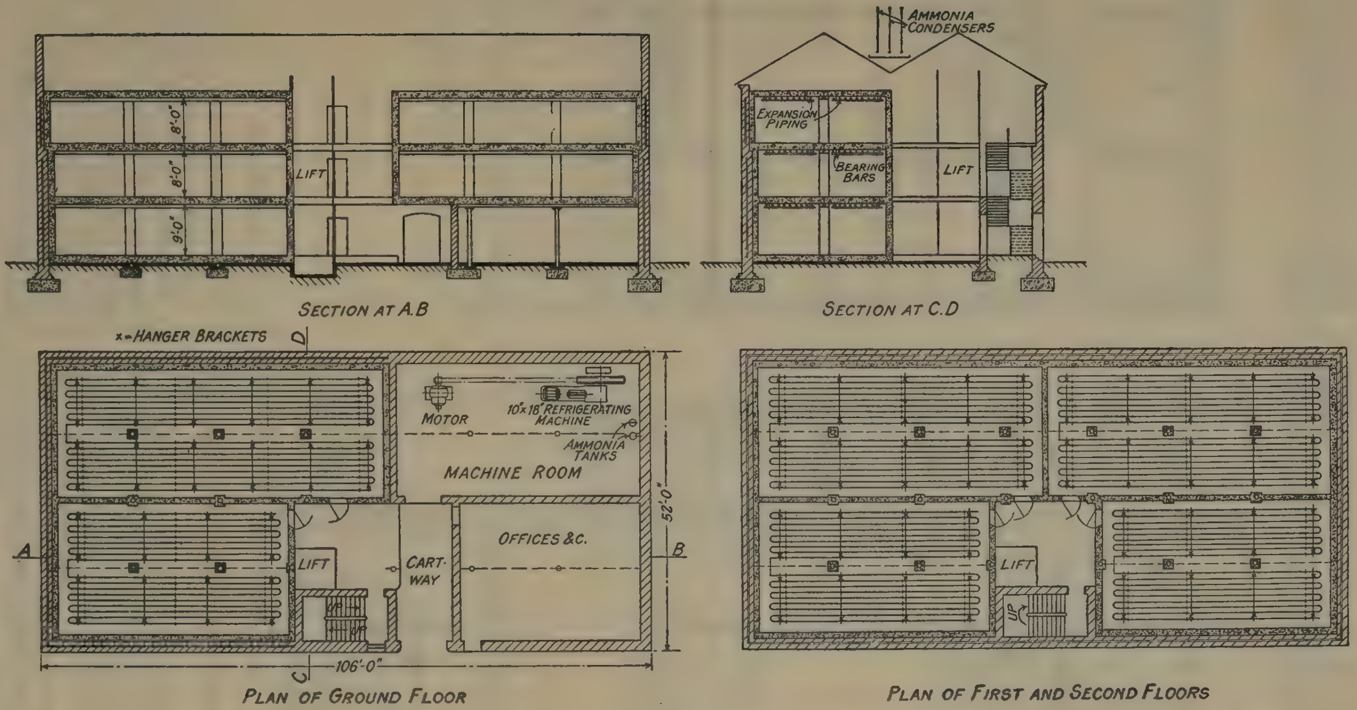


Fig. 28. Arrangement of Typical Small Cold Room

There are three methods by which this cooling is performed—

1. By direct expansion.
2. By brine circulation.
3. By cold air circulation.

In the first. flat grids of pipes are secured to the walls and ceilings of the cold store chambers, and the refrigerant, ammonia, or CO_2, is allowed to evaporate directly in these pipes; hence the name, direct expansion. This system is the cheapest in first cost and in operating cost whenever it is practicable. Cases arise, however,



SECTION AT A.B

SECTION AT C.D

PLAN OF GROUND FLOOR

PLAN OF FIRST AND SECOND FLOORS

Fig. 29. General Arrangement of Large Cold Store

when it is not desirable to use it, and it possesses the great disadvantage that immediately the machinery stops, the temperature in the cold chambers will begin to rise, more or less rapidly, depending upon the conditions.

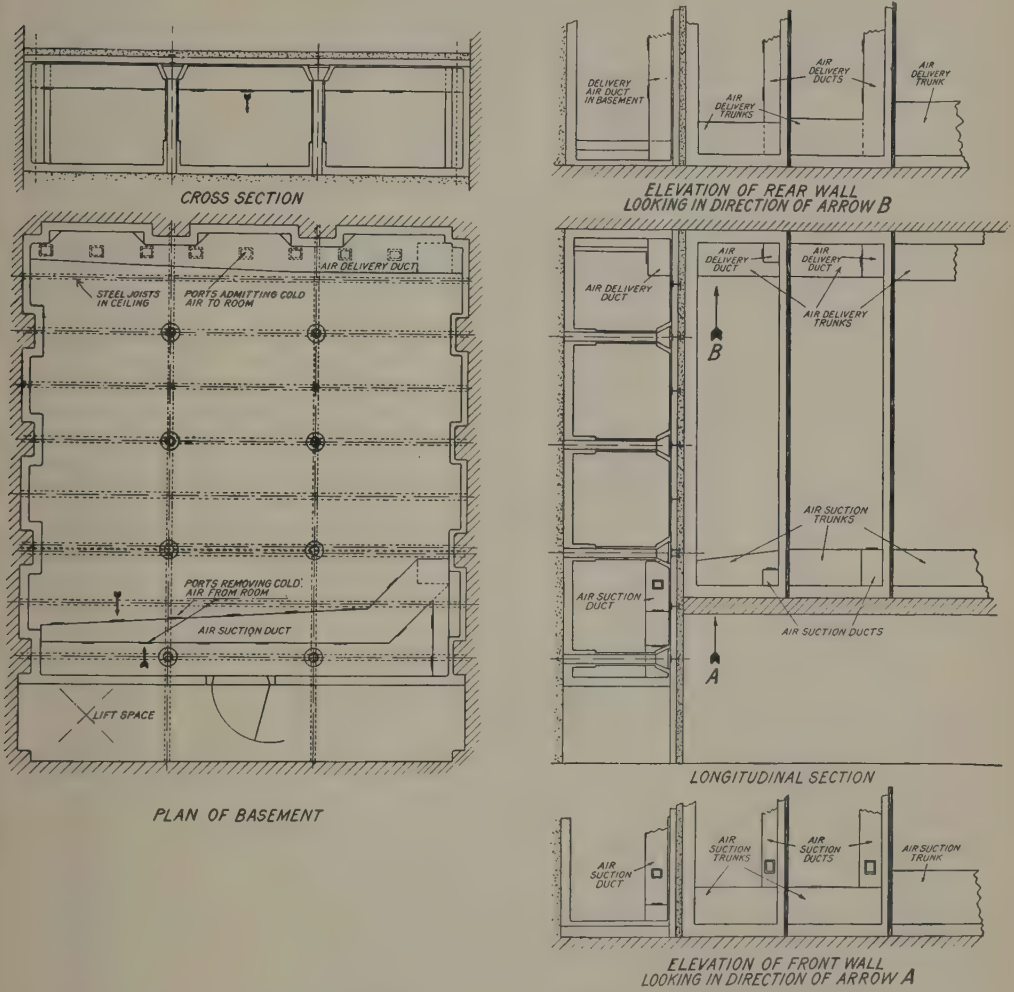


FIG. 30. ARRANGEMENT OF COLD ROOMS FOR COLD AIR CIRCULATION

In the second system, the refrigerant is employed to cool brine which is then pumped round the pipes fitted on the walls and ceilings. This system is more costly than the direct expansion system, because firstly, a set of coils or pipes is necessary by which the refrigerant cools the brine, and then a set of brine pipes is required for the cold rooms. There is thus a double set of cooling pipes required together with a double transmission of the cold, and this makes for increased operating cost.

With the brine system the temperature in the rooms will not rise rapidly when the machinery is stopped, because the great weight of cold brine in the pipes will itself keep the rooms cold for a considerable time.

While both the foregoing systems will keep the rooms cold, they fail in one essential item which is of vital importance in the successful storage of some kinds of goods, such as fruit, eggs, etc. This point is that of ventilation. Fruits, etc., give off gases during storage which, if allowed to accumulate in the rooms, would contaminate the goods. Neither the direct expansion nor the brine system provides for the removal of such gases nor indeed any ventilation at all, except such as accidentally occurs by the opening and closing of the doors. We therefore find that goods of the kinds just mentioned are cooled by some system of air circulation. With this there are often no pipes fitted in the cold rooms at all. The cold pipes are contained in a small chamber or casing and a fan blows air over them. The air thus cooled is led by a system of trunks or passage ways to the various cold rooms. In the room a trunk delivering the cold air is arranged to run the whole length of the room, while on the opposite side of the room there is a trunk connected to the fan suction and through which air is continually withdrawn from the room. The foul gases are thus continuously withdrawn, and arrangements are made to take in periodically some fresh air from the atmosphere, pass it over the cooling pipes and deliver it into the rooms. These are thus kept sweet. The pipes used for cooling the air are often cooled by brine, or brine is showered over the outside of them and collected in a tank underneath. This is very desirable, as it washes the air in circulation and gives a more effective cooling, and, further, the cold brine forms a reserve of cold by means of which the rooms may be kept cold after the refrigerating machine is stopped. This is done by keeping the fan running and circulating the air over the pipes down which the cold brine is running.

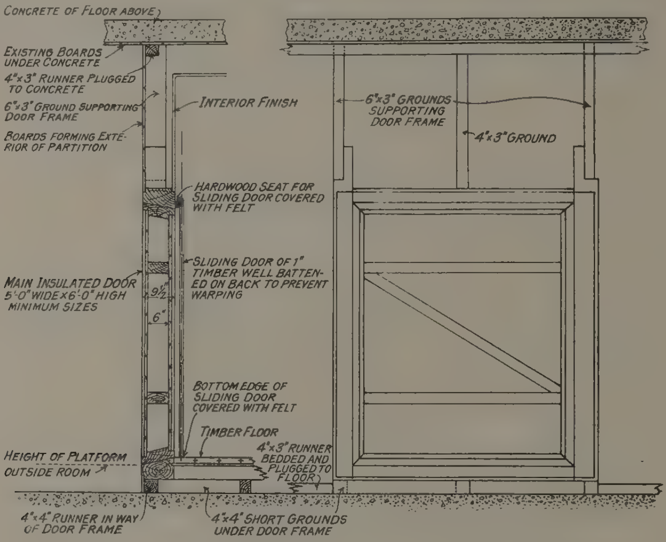

CONCRETE OF FLOOR ABOVE
EXISTING BOARDS UNDER CONCRETE
4"×3" RUNNER PLUGGED TO CONCRETE
6"×3" GROUND SUPPORTING DOOR FRAME
BOARDS FORMING EXTERIOR OF PARTITION
INTERIOR FINISH
HARDWOOD SEAT FOR SLIDING DOOR COVERED WITH FELT
SLIDING DOOR OF 1" TIMBER WELL BATTENED ON BACK TO PREVENT WARPING
MAIN INSULATED DOOR 5'-0" WIDE × 6'-0" HIGH MINIMUM SIZES
9½"
6"
BOTTOM EDGE OF SLIDING DOOR COVERED WITH FELT
TIMBER FLOOR
4"×3" RUNNER BEDDED AND PLUGGED TO FLOOR
HEIGHT OF PLATFORM OUTSIDE ROOM
4"×4" RUNNER IN WAY OF DOOR FRAME
4"×4" SHORT GROUNDS UNDER DOOR FRAME
6"×3" GROUNDS SUPPORTING DOOR FRAME
4"×3" GROUND

FIG. 30A. INSULATED DOOR FOR COLD STORAGE CHAMBER

Fig. 28 illustrates a typical small cold room for the storage of meat, and is such as would be used at a large butcher's shop, or hotel, etc. The cooling in this instance is done by direct expansion or cold brine pipes on the side walls. At one end of the room a screen or partition of timber is constructed, and behind this is placed a tank containing brine. Inside the tank is placed a grid or coil of pipe which is connected to the other cooling pipes on the walls. While the machine is working, the brine in this tank is cooled, and afterwards it serves to keep the room cool when the machine is stopped. In order to keep the atmosphere of the room crisp and dry, a small electrically-driven fan is fitted in the timber partition and circulates the air over the brine tank and through the room. Any excess moisture in the air is condensed out of it and deposited on the brine tank. Periodically, of course, the room must be emptied, and allowed to warm up in order to thaw off any snow and ice which may have formed on the piping.

The thawing-out has to be done at longer or shorter intervals with all rooms cooled by pipes, whether direct expansion or brine circulation and whether large or small. The frequency of thawing-out will depend upon the particular working conditions.

Fig. 29 shows the general arrangement of a larger cold store cooled by direct expansion pipes. The arrangement of the piping on the walls and the general arrangement of the stores, lifts, etc., will be noticed.

Fig. 30 shows the arrangement of a cold room cooled by air circulation, and shows the air trunks and ducts running along two sides of the room.

All cold rooms must be insulated to prevent heat leaking in from the outside and raising the temperature. In Fig. 28, the insulation consists of two layers of cork slabs secured to the walls, floor, and ceiling by cement or bitumen. The surface of the cork slabs inside the room is coated with cement plaster and painted or enamelled. The floors are finished with a coating of asphalt or granolithic. This is the best method of insulating such rooms, as it permits of their being kept perfectly clean. The older method of using granulated cork or similar material encased in boards is not so good. Apart from trouble due to the timber rotting, such insulation is liable to harbour vermin, and to acquire odours from strong smelling goods and transmit them to more delicate goods.

Fig. 30A shows the type of insulated door used in cold storage chambers. The framing and door posts are of heavy section pitch pine securely braced together. The interior of the door is filled up with cork insulation and faced with boards. All doors should have fasteners which can be operated from either side of the door. Hinges should be adjustable because these heavy doors drag on their hinges and require occasional adjustment. The sills of the doors are usually covered with a galvanized iron plate to prevent wear of the timber sill.

(Conclusion)

MACHINE TOOLS

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XXII

MULTIPLE SPINDLE AUTOMATICS

The Multiple Spindle Automatics. The designs of these are varied greatly. Some are built for bar work only, others—these often with vertical spindles—for chuck work only. Four to five working spindles are usually included, carried in one head, all in simultaneous action and wholly automatic, the rotation, the indexing, the gripping, the feeding going on continually, while the duties of the attendant are limited to supplying stock and removing finished pieces. The design is only adapted for work that is highly repetitive. When shop space is limited, a multi-spindle machine occupies no more floor area than a single spindle automatic does. Lengths of shafting, costs of supervision, and overhead charges are reduced. Against these, the higher outlay on the machines has to be set off. The question becomes that of the relation of output to expenditure.

Some of these machines have no turret, but its place is taken by tool-holding bars, which are set in alignment with the work-holding spindles, rotated and indexed. The tool bars take the place of the tool holders that are attached to turret faces. Each bar is equipped with its tool or set of tools as required for roughing and finishing, turning, boring, threading, etc. The tool equipment for producing a simple article may be very elaborate, yet when tens or hundreds of thousands of similar pieces are being made, the cost per piece, though involving the action of several tools is ridiculously small, being often only the fraction of a penny.

Some machines by the New Britain Machine

Company of New Britain, Connecticut, U.S.A., are of the chucking type. The spindles in the head, which is rotated and indexed, carry the revolving tools, while as many pieces of work as there are spindles are held stationary in the multiple chuck turret which advances and feeds the work against the revolving tools. The centres of the work-holding sections of the turret are arranged in a true polygon, around the axial centre. The centres of the tool spindles are aligned exactly with all but one of the turret section centres on the polygon. As the turret indexes it brings the chuck section opposite where there is no tool, where the operator is stationed for unloading and loading. In this machine the turret advances and feeds the work against the revolving tools. A double head machine is also made in which two sets of revolving tools advance from opposite sides to a central turret, so halving the time occupied when both ends of a piece have to be tooled.

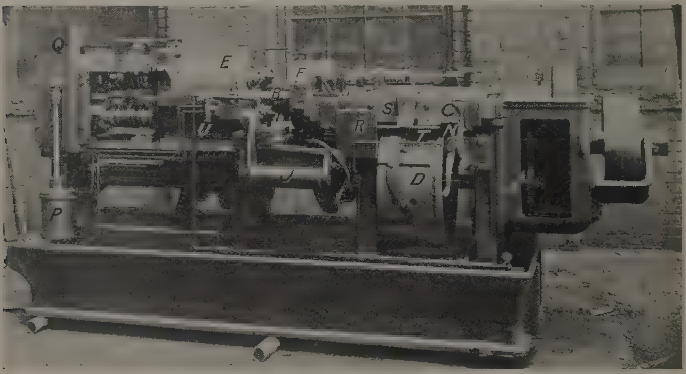


Fig. 156. Multiple Spindle Automatic with Air Control

The most recent single head machine of the work rotating type is shown by the photograph, Fig. 156. The novel feature is an air control operated by a single valve each time a work spindle is indexed into the loading position. The object is to relieve the operator of the labour of hand chucking, giving him time to attend to two machines. The control disengages the drive to that particular spindle, applies a brake to stop the rotation quickly and actuates the clutch mechanism to release the work. When the chuck has been reloaded the air valve is operated a second time to grip the jaws on the work, release the brake and engage the drive to the spindle.

The machine shown has four work spindles, or six if desired. There are four tool stations on the longitudinal tool slide, one station on the front cross slide, and two stations on the rear cross slide, and each can be used separately or in conjunction. All movements are derived from the single speed pulley, seen at the right of Fig. 156, belt or motor driven. The drive is transmitted and varied through change gears in the box *A* to a shaft *B* that extends forward into the headstock to drive the tool spindles. The speeds can be changed without altering rates of feed, and vice versa. At the farther end of the shaft *B* a spur gear is mounted, that meshes with a pinion on the spindles which are disposed in a circle round the shaft, and contained in a cylindrical casing that has bearings in the housing *E*. A spur gear of large diameter is mounted directly on the spindle cylinder to provide for the indexing movements. From a second set of change gears in the box *A*, and gears controlled by a clutch handle *C* to drive the cam drum *D* by means of an internal ring gear. The drum drives the shaft on which it is mounted, which extends to the forward end of the machine. At a point below the headstock a pair of elliptical spur gears, with Geneva mechanism, intermittently engages the slots of a member mounted loosely on the shaft, and geared to the indexing ring on the spindle cylinder, with the result that the indexing is accomplished during a brief portion of the rotation of the cam drum *D*. Round the spindle cylinder are slots equal in number to the spindles which are successively engaged by a locking bolt, withdrawn with a cam and lever, and thrust into place with springs.

As the work in this machine is carried in the headstock spindles the feeds are imparted to the tool slide *F*, the feed and return being imparted by the strips on the cam drum *D*, with a connecting shoe that can be attached to the slide in two positions for adjustment of the forward and return positions of the slide. On the camshaft driven by *D*, a disc with cams mounted on both sides controls the movements of the cross slides, that for the front slide being controlled with the lever arm *J*, a similar one being mounted at the back for the rear slide. There is a fast return drive to the cam drum, effected through gearing, to reduce the time of a cycle.

The air valve for controlling the spindle in the loading position is seen at *U*. As a finished part reaches the loading position the valve is operated to admit air into the upper portion of the cylinder *P* to push down the piston, and, through the connection, the slide *Q*. Two cam paths near the top of this slide operate successively the bars that control the spindle driving clutch and brake, and the chuck jaw operating rod and release. A third cam path on slide *Q* operates longitudinally a bar that extends towards the driving end of the machine, and is connected through a gear segment under the cover *R* to a bar *S* on which a dog is mounted. This disconnects the drive to the tool slides and the spindle indexing mechanism, in case the operator has not chucked a new piece of work by the time that the main tool slide has completed its return movement. Attached to this tool slide is a bracket to which is fastened the rod *T*. When the spindle has been reloaded the rod *S* and its dog is moved forward, the dog pushes the part containing the rod *T* sideways, and so the rod will strike the handle *C*, and stop the machine with the exception of the spindle rotation.

The Gridley Design. One of these is shown by Fig. 157. It has four spindles, for bar work, and is fully automatic, so that one man can attend to three or four machines. One of the characteristic features is, that the place of the turret is taken by a four-sided tool slide, with chamfered edges against which the tool holders are bolted. The advantage gained is that the tools are supported close to their cutting points with the avoidance of spring. All the tools are working simultaneously on the four stock bars, and they are cammed so that the different operations finish at the same time, including the cutting off of the finished piece. The alignment of the tool slide with the spindle-carrying cylinder is assured by the extension of the central part of the cylinder to the farther end of the machine, and mounting the tool slide on it. This extension is the spindle carrier, which has a rotating step-by-step movement, to bring the work spindles into alignment with the tools held on the tool slide, which does not rotate. After each partial revolution of the spindle carrier, with the bars of stock, the tools are moved forward, and each tool performs its required operation. Then the tool is withdrawn, the spindle indexed another step, so bringing the work into alignment with another tool. A finished piece is produced every time the tool slide moves forward.

The driving shaft at the centre of the spindle-carrying cylinder has a gear that meshes with a gear on each spindle. The shaft is speeded by change gears driven from a pulley running at a constant speed, or a motor. The spindles have the usual collets and fingers. During the indexing the start and stop are without shock, but the intermediate movement is rapid.

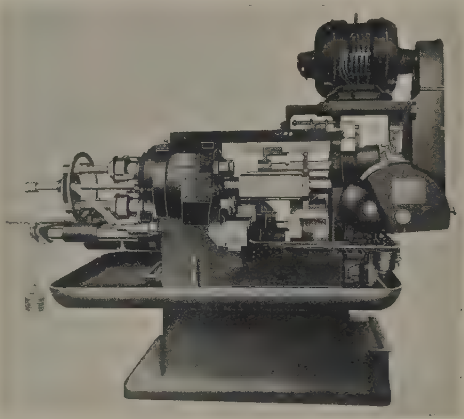


FIG. 157. FOUR-SPINDLE GRIDLEY AUTOMATIC

Movements are effected with cams on a lower shaft actuated from the main driving shaft through change and worm gears. It carries cams for feeding the tool slide, for operating the cutting off and forming tools, for operating the chuck and feeding stock mechanism which advances the bars while the tools are idle, and the mechanism for revolving the spindle cylinder.

It is not necessary to give attention to the feed when setting up the machine. The spindle and feed box are geared together, and the feeds are controlled by a lever which will feed a definite length independently of the number of revolutions of the spindle per minute. If the tools advance 1 in. while the spindle makes 125 revolutions, this is not so economical as the same advance with the spindle making 100 revolutions. The advantage of the design is, that the operator ascertains by tentative settings of the feed box lever the mean between a feed too fine and one too coarse.

Besides turning, the machine includes some valuable accessories for threading, reaming, and high speed drilling. Gear ratios are provided for threading with a cam through spur gears and a clutch, the ratios being related to the spindle speeds. Slow motions are provided for cutting; rapid, for running the dies off. Right- or left-hand threads can be cut. An accelerated reaming attachment permits of reaming a hole of greater depth than the travel of the tool slide. A high speed drilling attachment permits of rotating the drill into the rotating work, so rendering high cutting speeds available. The cross slide carries forming and cutting-off tools that are fed with cams at right angles with those on the longitudinal tool slide. Heavy cutting is divided between an upper and a lower tool, the first roughing, the second finishing.

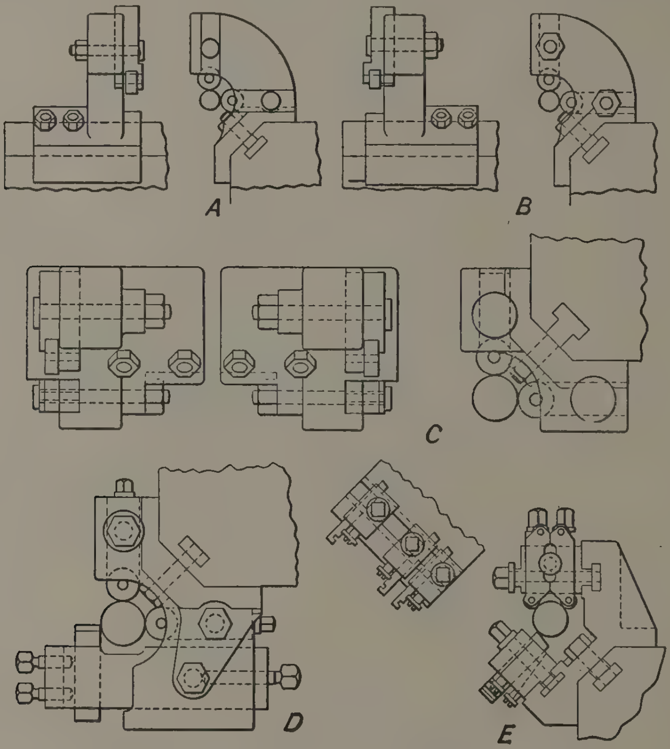


Fig. 158. Diagram of Roller Back-rest and Turners

The tools used are mostly of a simple, single-edged type. They are mounted singly, or in combination in holders that are straddled over the edges of the tool slide and secured with tee-head bolts. A group of turning tools is shown by Fig. 158. *A*, *B*, and *C* are independent back rests, used when heavy turning or forming is being done from the cross slides or the tool bar. *D* is with tool-holder. *E* shows three turning tools, individually adjusted with a collar head screw and opposed with back rests adjustable along the holder. In this design, more than one diameter can be produced at one traverse of the tool slide.

In all this class of machines, when a spindle is run at a definite speed, and more than one tool is in operation, it seldom happens that all the tools can be cutting at their most appropriate speeds. Then the time is set for the tool that performs the longest single operation.

(Conclusion)

DIESEL ENGINES
CONSTRUCTION, OPERATION, AND MAINTENANCE

By A. Orton, A.M.I.Mech.E.

LESSON XIII
GENERAL REVIEW

In this the last lesson of the present series, it is proposed to refer, necessarily very briefly, to the performance of present-day Diesel engines, the general relationship between Diesel engines and other modern oil engines, the state of development at which the Diesel engine has now arrived and its prospects for the future.

Diesel Engine Performance. Naturally, the greatest importance in this respect is attached to fuel economy, or rather heat economy. In the first lesson a comparison was made between the efficiencies of the Diesel engine and of some other types of engine. Virtually, the figures there given stand good to-day, except that the early low compression type of oil engines there quoted is now obsolete, and that some improvement has been made in the fuel consumption of the Diesel. It is now fairly common for the best Diesel engines, of both air injection and solid injection types, to attain fuel consumptions of less than 0·4 lb. of oil per brake horse-power hour at full power. Taking this figure and assuming a net heat value of the oil as 18,000 B.Th.U.'s per lb., we find that the thermal efficiency of the engine is 35·4 per cent on the brake horse-power basis, whilst on the indicated horse-power basis, assuming a mechanical efficiency of 78 per cent, the figure works out as 45·3 per cent.

Now the laws of thermodynamics show that the Diesel engine cycle, even if carried out in a theoretically perfect manner, that is without any heat losses except to the exhaust, could only attain an indicated thermal efficiency of about 59 per cent. Practical considerations, however, place the maximum *attainable* efficiency so much nearer 45 per cent that there is scarcely any hope of much further improvement in this respect. The only possibilities lie in further increasing the compression and maximum combustion pressures, or, by means of improved mechanical construction, increasing the mechanical efficiency. Little can be expected in either of these two directions, however, and though makers are constantly effecting improvements in detail, these are of importance chiefly from the point of view of reliability in running, convenience in operation, or cheapness in construction.

Typical performance curves for a good average four-stroke, constant speed engine are given in Fig. 42, the figures for the fuel consumption being taken on a comparatively conservative basis. From these curves the various characteristics associated with the changes from "no-load" to "full-load" can be studied.

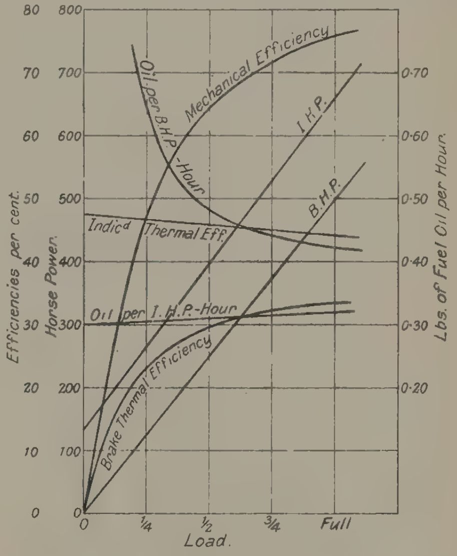


Fig. 42. Performance Curves for Four-stroke Engine

Present-day Oil Engines. We are concerned here only with engines which use heavy petroleum fuel oils, i.e. oils having specific gravities not less than ·85, and there are two main types which call for any special comparison with the Diesel engine, the so-called semi-Diesel engine and the high compression cold-starting engine respectively.

The first is an engine known variously as the "hot bulb" or "surface ignition" type, because the essential principle of its working is that the fuel oil is sprayed into the cylinder so that it impinges upon an uncooled, and therefore very hot, portion of the combustion chamber walls. The heat of this surface is the main factor in bringing about the vaporization and ignition of the oil, the compression pressure, and consequently the compression temperature, being too low to effect this of itself. Owing to this low compression, usually from 150 to 180 lb. per sq. in., it is impossible to start the engine until the hot bulb has been artificially heated to such a degree that it will ignite the fuel oil when sprayed upon it. The engine is, of course, set in motion in the usual way, by hand (if small enough) or by compressed air, and for a short time the hot bulb is kept heated by a blow-lamp or burner, but afterwards is maintained hot by the heat of combustion in the cylinder. Ignition of the fuel then takes place automatically as the combined result of the hot surface and the compression temperature. The maximum combustion pressure in these engines amounts to about 300 or 350 lb. per sq. in.

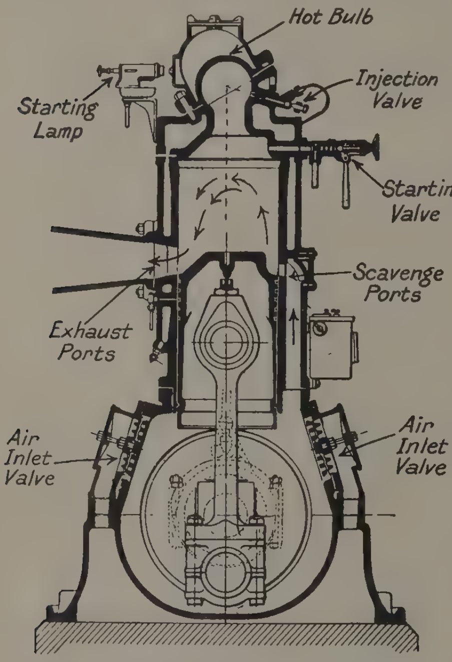


Fig. 43. Typical Hot Bulb or Semi-Diesel Engine with Crank Chamber Scavenging

Engines working on this principle are made both on the four-stroke cycle and the two-stroke cycle, the former more often as horizontal engines and the latter as vertical. The most popular type at the present time is the latter, a vertical, two-stroke, port-scavenging engine, in which the underside of the main piston acts as a scavenge piston, and the crank chamber itself, or in some cases a space above it separated from the crank chamber by a diaphragm, serves as the scavenge air chamber. Fig. 43 is a cross-section of a typical engine of this class, and shows the general construction quite clearly. This is, perhaps, the simplest form of oil engine made and, for certain purposes, is very suitable. But its dependence for ignition on a hot surface, the heat of which varies with the load on the engine, renders it somewhat sensitive to, and less able to deal with rapid changes of load, and its inherently lower fuel economy place it on a definitely lower plane of technical excellence than that of the Diesel engine.

The second main type, known as the "cold-starting high compression" engine, is, in all essential respects, a Diesel engine with solid injection. In fact, on the Continent and in America, such engines are almost invariably described as "compressorless" Diesel engines or "airless injection" Diesel engines. The compression pressure of such engines is usually equal to, or nearly equal to, that of a normal or "full" Diesel, while the type of combustion and the pressures attained correspond essentially with those already described in previous lessons for the solid injection Diesel engine. These engines are chiefly manufactured in the horizontal and vertical four-stroke types, but to some extent also as vertical, two-stroke port-scavenging engines. In so far as they are successful in attaining a similar degree of reliability they possess all the advantages, which have already been referred to, associated with the solid injection system, that is, the absence of the cooling effect of the blast air and the necessity for its regulation and the avoidance of any continuously running air compressor. They naturally possess also its disadvantages.

Between these two main types there are also intermediate types employing medium degrees of compression, about 300 lb. per sq. in., having fully water-cooled combustion chambers, but employing some form of auxiliary apparatus to get over the initial difficulty of starting when cold. One such form of apparatus is an electrically-heated plug or coil fitted into the combustion space, and which is rendered incandescent before starting by an electric current from a battery or other convenient source, thus forming a "hot spot" which causes the ignition of the oil spray at the first injection. When the engine is properly running the electric current is cut off, and the heat of compression is then sufficient to ensure regular ignition. Other means of effecting the same purpose are adopted also, such as, in small engines, the insertion of a glowing cartridge of paper which burns completely away when the engine is running, or, as in certain large marine engines, the heating up of the cylinders before starting by passing steam or hot water through the engine jackets.

For each of these various types of oil engine some particular advantage can be claimed, such as the simplicity of construction, the low maximum pressure and cheapness of the hot bulb two-stroke engine and, to some extent, also of the medium compression engines of the same type, which can claim in addition a fuel economy approaching that of the Diesel engine. Yet, in so far as any of them fall short of the full Diesel principle of high compression and self-ignition, so must they suffer some inherent disadvantages, such as lower economy, inability to deal so well with rapid changes of load, and the necessity of using extraneous devices for starting. The principles of the Diesel cycle are such that the engine is able to meet all conditions of load, within its proper capacity, equally well, because the factors controlling ignition and combustion are independent of any changes that depend on the load.

Thus it may be claimed that the Diesel engine constitutes the highest standard of perfection towards which all other types tend to converge, and this view is supported by the present tendency of oil engine design generally. Low compression is everywhere being superseded by high compression, and with it are disappearing all those devices for starting or for meeting unfavourable changes of load which have been necessary in the past. It might well be predicted, therefore, that the ultimate form of the reciprocating heavy oil engine will be not far different in any essential principle from that of the present Diesel engine.

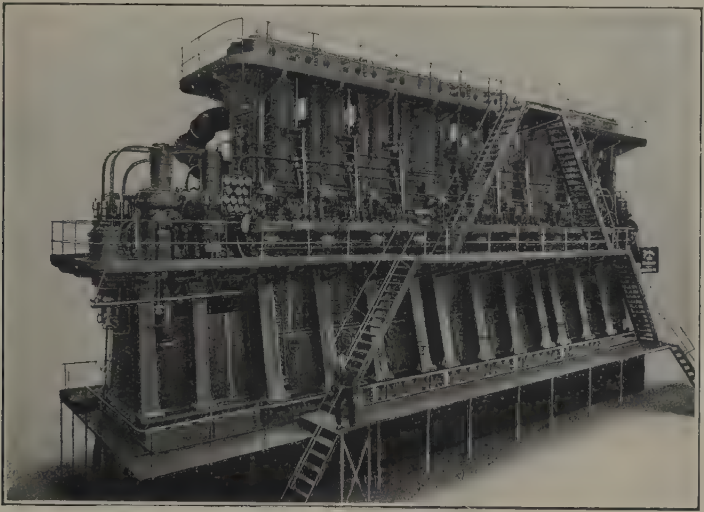


FIG. 44. 15,000 B.H.P. DOUBLE-ACTING DIESEL ENGINE

Present Position and Prospects of the Diesel Engine. The commercial development of the Diesel engine may be said to have begun with the present century. Before the year 1900 comparatively few engines existed, and none of more than about 100 b.h.p. Progress at first was slow, but since about 1908, apart from the years of the war, developments have taken place so rapidly that the engine is now recognized as a most reliable prime mover, suitable for both small and large units, and withal the most economical one thermally that is available. It has proved itself to be eminently suitable for the exacting requirements of electric power generation on land, and has also been adopted so largely and successfully for ship propulsion that, in the year 1925, no less than 36 ships, each of not less than 2,000 tons dead weight, and totally over 321,000 tons dead weight, were completed in British shipyards alone, the total output for the world in the same period being 127 ships, totalling over one million tons dead weight.

It has been impossible for reasons of space to refer in the present articles to any of the special apparatus necessary to adapt the Diesel engine for marine purposes, but it may be stated simply that no insuperable difficulties have been met in fulfilling all the demands of the case. The problems associated with the reversing of marine engines rapidly and certainly, for long-continued manoeuvring operations, have been completely solved and, in this respect, the modern Diesel marine engine is in no way inferior to the best steam-engines.

As regards the powers available the position may be indicated by the fact that there are now ships at sea fitted with British built propelling engines of 10,000 i.h.p. in one unit, these engines being of the double-acting, four-stroke type, with six cylinders, and constituting the largest yet put into this service. On land and abroad equally remarkable progress is being made, for recently a Diesel engine of 15,000 b.h.p. was installed in a German power station, this engine having nine cylinders and working on the double-acting, two-stroke cycle, and forming the largest single unit yet built. Similar engines will also be available for marine purposes, so that within a comparatively short period the power requirements of even the greatest liners will be met. As a fitting conclusion to this brief statement we give, in Fig. 44, an illustration of the above engine, and it may be of interest to mention that its cylinders are 33·8 in. in diameter, with a stroke of 59 in., whilst the overall height is nearly 39 ft. and the length 76 ft.

Great constructional difficulties have had to be overcome in bringing the Diesel engine to its present stages of development and there are still, of course, many more ahead. But in view of the ever increasing knowledge that comes of scientific research and practical experience, resulting in improved and more suitable materials of construction and their better application, it is certain that these also will in due course be surmounted, and that the Diesel engine will continue to extend its spheres of usefulness, of which perhaps the most important, and the one in which, by its high thermal efficiency, the Diesel engine most distinguishes itself, is the conservation of the world's supply of fuel.

(CONCLUSION)

POWER TRANSMISSION BY CHAINS

By H. M. Andrew, Wh.Ex.

LESSON V

SOME EXISTING APPLICATIONS OF CHAIN DRIVE

Pedal Cycle Drive. The earliest important application of driving chain was its adoption by the pedal cycle. It seems strange that in an application such as this, where easy running and good efficiency so directly tend to the benefit of the user, the chain case is the exception rather than the rule, and lubrication is generally either sadly neglected or omitted altogether. Where both are in regular use the beneficial effect is at once realized in the long life and sweet running of the drive.

The larger the chain wheels the better, although the diameter of these is usually restricted in practice by the back fork and the designer's sense of proportion.

Motor-cycle Drives. Engine speeds have increased so considerably of late, particularly in speed models, that the engine to countershaft drive tends to far exceed the desirable speed for the chain usually employed. In order not to reduce the number of teeth on the wheels it seems desirable to use the smallest possible pitch in preferably a duplex type, for this drive. Some makers use a gear drive from the engine to countershaft, although it is questionable whether any better mechanical efficiency is obtained thereby.

The rear drive, at, say, half the speed of the front drive, is an eminently suitable case for chain drive, and the chain wheels should be the largest practicable consistent with clearing the back fork and giving sufficient ground clearance. The problem of enclosing and lubricating this chain is a difficult one, but an underguard at least is desirable, and some form of lubrication should be used regularly, even if it only consists of soaking the chain at intervals in a graphite grease.

Rear Axle Drives on Motor Vehicles, Locomotives, etc. In the majority of these drives the ratio of speed reduction employed and the necessity of providing adequate ground clearance for the rear chain wheel has led to an unfortunate use of pinions with very few teeth. Coupled with frequent omission of chain cases and serious neglect of lubrication, the drive is usually far from ideal. Much, however, is now being done in some designs to improve matters.

A lower speed ratio, where possible, giving a larger pinion, would improve matters considerably, and at least an underguard, to keep stones and dirt out of the chain, is essential, together with periodic, if not continuous, lubrication.

Motor-car Front End Drives. Inverted tooth chain is largely employed for these drives, usually in the form of a three-wheel drive from the crankshaft to the camshaft and magneto or the camshaft and dynamo. The third wheel driving the magneto or dynamo is generally made adjustable. Sometimes a two-wheel adjustable drive to one auxiliary supplements the three-wheel drive. The early style of using non-adjustable two-wheel drives is now largely superseded by the three-wheel drive with adjustment, as this enables a fuller use of the chain to be obtained.

The housing and lubrication of these drives is usually excellent. The life of drive obtained in practice is rarely the full 3 per cent of wear available in the chain bearings as, after about half this wear, the cyclic irregularity inevitably present in cam drives tends to cause noise in the drive and, owing to the high degree of quietness called for, it is often customary to renew the chain at that stage.

Propeller Drives. Air propellers on hydro-gliders and similar craft usually employ bush roller chain at fairly high speed, as a light yet robust drive is required, and any noise in such drives is generally negligible in comparison with the hum of the propellers. A chain case is generally omitted on the ground that it would interfere with the air currents, and this prevents the use of adequate lubrication suitable to the high speed employed.

For short runs, chains soaked in a good graphite lubricant may serve the purpose, but for longer runs the chain might be enclosed in stream lined tubes coupled to wheel boxes, with oil lubrication.

Impulsive Drives. In driving air compressors, punching and shearing machines and some types of pumps and fans, there is often a fair degree of cyclic irregularity accompanied by shock load or jar on the transmission. The fly-wheel usually employed with such machinery should be sufficiently heavy to prevent more than, say, 2 per cent speed variation in the transmission, and the driving chain should be kept closely adjusted, with a minimum of slack, to prevent snatching and flapping.

Inverted tooth chain is often used in compressor drives to secure silent running, but the bush roller type is a preferable design from all other aspects for drives of this type.

Lineshaft Drives from Motor on Slide Rails. As lineshafts are usually rather low speed it is all to the good in these cases to use chain wheels with the largest possible number of teeth that the required ratio of reduction will permit, without exceeding the desirable limit laid down in Lesson I.

Protection and lubrication in such drives is usually excellent, and very satisfactory service is obtained in the majority of cases.

Where one wheel is larger than the other in drives inclined to the horizontal, the larger wheel should preferably be on top, and it is better to have the upper chain line as the driving or tight side if choice is possible.

Chain drive has been used with great advantage to replace the vertical shafts and bevel gear drives formerly employed in many textile mills, etc. Considerable saving in power loss is usually obtained in such cases.

Many textile mills use inverted tooth chain driving from motors to main lineshafts, and bush roller chain from main shaft to countershafts. A very even drive is obtainable by driving chain that ensures uniformity of product.

Paper Mill Drives and Calendering Processes. The manufacture of paper and certain fabrics often demands very smooth and even movement of drums, and the quality of the product is appreciably affected by speed variations or irregular slip. Chain drive is extensively used and appears to give excellent results.

Bush roller chain can be threaded in and out of a series of calendering drums to drive them in a similar manner to a train of gears, producing frictional action between drums if required by variation in the number of chain wheel teeth. Chain has some advantage over gears in such a drive, as the chain retains its grip when the drums are separated a little by material passed between them, whereas gears would develop backlash.

Loom Drives. Chain drive is now used extensively where looms are individually driven, both bush roller and inverted tooth types being employed for this purpose. A frictional slip clutch is often employed on one chain wheel to knock off the loom in case of mishap. The driving motor can be made adjustable by swinging it on a side pivot with its chain wheel hanging in the light of the driving chain. An outer pillar supporting the free side of the motor is made adjustable.

Drives in Confined Spaces. There are a great many cases where fairly short drives are required for which a train of gears is inadmissible or too expensive to install, while the available space is too small for a satisfactory belt or rope drive. Such cases provide the natural field for chain drive.

Drives in underground workings, in ships, and in confined workrooms, are typical cases. Driving chain offers a very suitable method of transmitting power in such cases and, with a little care in installation and a moderate amount of attention, very satisfactory results are obtainable.

It may be desirable in concluding these articles to recapitulate the guiding principles in selection.

1. Do not design for the smallest permissible chain pinion. Always think in terms of the largest number of teeth you can afford to install.
2. Protect the chain adequately from dust and dirt and, if the speed is high, always use an oil bath chain case.
3. Provide ample lubrication. An excess is not wasteful, as it can be collected, filtered, and used again.
4. Always have the chain wheels cut with the correct cutter specified by the chain maker.
5. Take care in the initial alignment of the drive and attend regularly to adjustment.

(CONCLUSION)

STRUCTURAL STEELWORK

By W. Arnold Green, M.A., B.Sc., A.M.Inst.C.E., M.I.Struct.E.

LESSON VIII

ROOF TRUSSES

The simplest form of ridged roof truss consists of two inclined rafters connected at the apex and resting on two walls (Fig. 38(*a*)). Each rafter may be regarded as an inclined beam with vertical reactions at each end. If the joint at the apex is equivalent to a pin connection the vertical reaction at the apex must be resisted by axial thrusts in the rafters, which will bring outward thrusts as well as vertical loads on to the walls.

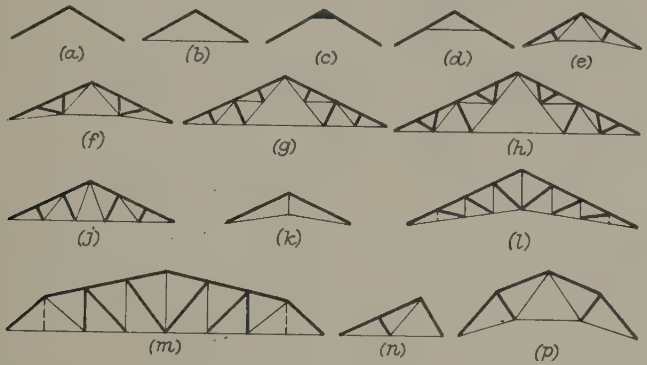


Fig. 38

To take the horizontal thrust, a tie bar can be inserted as indicated in Fig. 38(*b*), and the loads on the wall for vertical loading on the roof will be vertical, except that the tie bar will stretch slightly and the supports will tend to be pushed apart.

If the apex joint can be made sufficiently stiff to resist the central bending moment, due to vertical loads and end reactions, the tie bar would be unnecessary (Fig. 38(*c*)), and the only horizontal thrust on the walls from vertical loads would be due to the bending of the rafters and joint. The horizontal thrust can be avoided by fixing one end and placing the other end on rollers.

Instead of the stiff joint indicated in Fig. 38(*c*), a tie bar could be inserted between the apex and the shoe as in Fig. 38(*d*). The horizontal movement apart of the shoes will then be due to the combined bending of the rafters and stretch of the tie rod.

For the rafter beam shown in Fig. 38(*d*), a trussed rafter may be substituted as shown in Fig. 38(*e*), which illustrates a cambered *Fink* truss and Fig. 38(*f*) a cambered *Fan* truss. The bottom ties are often made horizontal as in Fig. 38 (*g* and *h*) which show a compound Fink and Fan truss respectively.

Another form of bracing is shown in Fig. 38(*j*), where the struts are at right angles to the rafter.

The bottom tie of the truss shown in Fig. 38(*b*) may be cambered as shown in Fig. 38(*k*), in which case a vertical tie bar must be connected to the apex.

The rafter of this type of truss may be strutted as shown in Fig. 38(*l*), where all the ties are vertical. The bottom ties in these types too may be horizontal, but this results in longer struts though, of course, the loads on the main ties and rafters are less.

Vertical sag rods are sometimes introduced as indicated by dotted lines, but these take no stress from external loads unless a load, such as a ceiling, is suspended from the bottom chord.

Fig. 38(*m*) shows a *Pratt* truss. The first vertical is a sag rod, the second is a strut and the central vertical is a tie.

The slope of the rafters may be different on each side of the apex as in the case of a north light roof truss (Fig. 38(*n*)), where the short side is glazed and faces the northern sky.

Fig. 38(*p*) shows a *mansard* roof truss.

For light trusses carrying corrugated sheeting, the rafter is often curved to a radius and sometimes the curved corrugated sheeting itself forms the rafter.

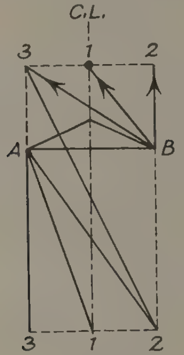


FIG. 39

The axial stresses in the various members are usually found graphically by reciprocal stress diagrams dealt with elsewhere.

For vertical loads, only half a symmetrical truss need be analysed. For a wind load taken as normal to the slope on one side, if the truss is fastened to walls or stanchions at each shoe, it is common to assume that the horizontal component of the wind load is divided equally between the two supports, in which case, if in Fig. 39 the full line 32 represents the total wind load in magnitude and direction, the reactions are represented by the lines $B1$ and $1A$. If the truss is on rollers at B, the reaction must be vertical ($B2$), and the reaction at A is represented by the line $2A$. If the rollers are at A, the reactions are represented by $B3$ and $3A$.

Fig. 40 shows the general arrangement of a steel roof framework. It is usual to make two adjacent trusses at the ends into a stable frame by cross bracing. The other trusses are strutted from this frame by the purlins. In a long roof there may be several of these braced frames.

The bottom ties are usually connected by light members parallel to the ridge. Where there is horizontal bracing in the plane of the ties as indicated by dotted diagonal lines in the figure, the ties must be of a section that will withstand compression.

It is often specified that all the members of of a truss must be able to act as struts. In the case of angles ties connected by rivets through one flange, as the load is applied eccentrically, it is usual to neglect half the area of the outstanding leg to compensate for stresses due to bending.

When a very stiff roof is required, it is sometimes necessary to put in bracing in a vertical plane, in which case the truss is made with vertical members as illustrated in Fig. 38(*l*) and (*m*), and a vertical bracing girder put in the plane of two or more of them.

By an experienced electric welder joints in a roof truss can be made with greater stiffness and with less metal than with riveted joints. As the sectional area of the members is not reduced by holing, a further economy results; but the practice of electric welding of structures is not yet common. A few riveted joints, in a truss built of angles and plates, are illustrated in Fig. 41.

The stiffness of the joints is neglected in determining the axial stresses in the members, which are considered as pin connected.

When the truss deflects under load, the stiffness of the joints restrains the free movement there would be in pin connected structure and stresses are set up at the joints. An attempt is rarely made to estimate them, as the design stresses are usually sufficiently low to allow for increases due to *secondary* stresses as they are called.

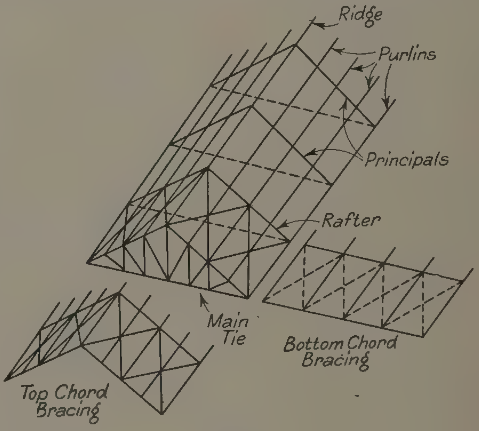


FIG. 40

These joint stresses are considerably increased if the lines of action of the connected members do not meet in a point, and care should be taken to reduce such stresses to a minimum. Truss shoes, for instance, are often made as illustrated in Fig. 42, but it is not difficult to ensure that the lines of rafter thrust and tie tension intersect above the centre line of the reaction, as shown in Fig. 41.

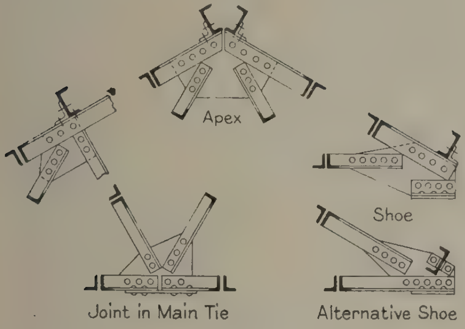


FIG. 41

The slope of the rafters depends on the nature of the covering used, as it is important that water will drain off without the risk of being blown back at the joints. Thus, the minimum rise of roof to the span is given in Hurst's handbook as 1/10 for corrugated sheets, 1/5 for slates (Queen's), 1/4 for ordinary slates, 1/3 for pantiles, and 1/2 for plain tiles.

For fixing wood sheeting for slates or tiles, wooden purlins are often employed. These are jointed over the trusses and secured by bolting to angle cleats riveted to the rafters. Fixing strips of wood may be bolted to the flange or web of joist or channel purlins, and the sheeting nailed to them instead.

Sometimes cement tiles are laid direct on steel channel or angle purlins remaining in position by their own weight (see Fig. 43).

To prevent lateral movement of the purlins, which are usually not stiff in the direction parallel to the rafter, they may be tied up to the ridge purlins by short round bars screwed at each end, passing through holes in the web and secured by nuts.

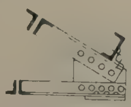


FIG. 42

A few typical purlin connections to rafters are shown in Fig. 44, and to sheeting and glazing in Fig. 45. It should be noted that the angle section is much stiffer for carrying vertical load if placed as in Fig. 43, though in that case sheeting has to be fixed by a special hook bolt, or a strap right round the angle.

It is advisable if possible to arrange the purlins so that they come at the panel points and produce no bending in the truss rafter. When it is necessary to place the purlins between the panel points, the bending moment produces stresses which necessitate a larger rafter section.

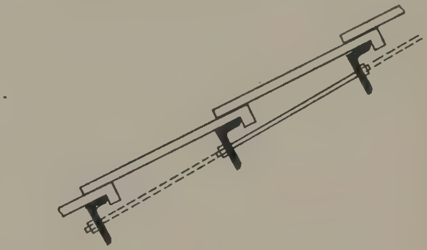


FIG. 43

The effective length of the strut for bending in a plane at right angles to the truss is, however, lessened.

If it were required to test the adequacy of a rafter consisting of two 5 in. × 3 in. × $\frac{5}{16}$ in. angles $\frac{1}{2}$ in. apart and 10 ft. between panel points, carrying a normal load of ·8 tons due to vertical loads and ·9 tons due to wind, applied 2 ft. 6 in. from one panel point, and also an axial load of 9 tons due to vertical load and 4 tons due to wind, and if B.E.S.A., No. 153, stressses were specified with a 25 per cent

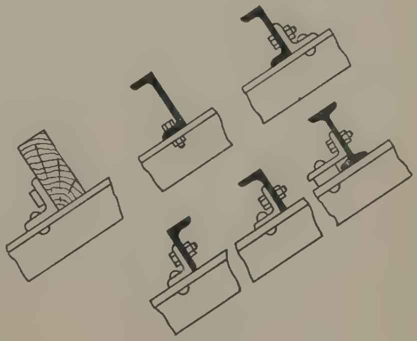


FIG. 44

increase for total load including wind, the calculations might be made as follows—

For two 5 in. × 3 in. × $\frac{5}{16}$ in. angles $\frac{1}{2}$ in. apart with short legs on top or horizontal (see Fig. 45) $k_x = 1{\cdot}6$, $k_y = 1{\cdot}24$, $A = 2 \times 2{\cdot}4$, $Ix = 2 \times 6{\cdot}14$, $n_c =$ distance of centroid from top $= 1{\cdot}66$. As in each case, the wind load is more than 25 per cent of the vertical load, the rafter must be designed for the total compression of 13 tons, with a bending moment due to 1·7 tons, 2 ft. 6 in. from end.

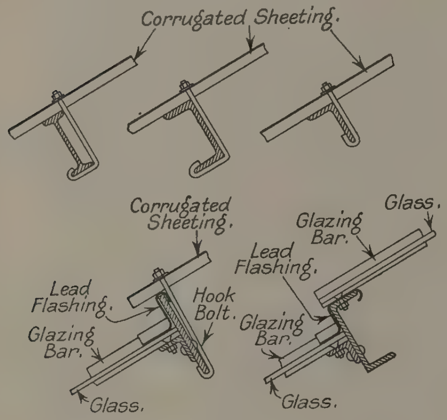


FIG. 45

The free bending moment is $1{\cdot}7 \times 2\frac{1}{2} \times 7\frac{1}{2} \times 12 \div 10 = 38{\cdot}2$ in. tons. As there is restraint at each end of the member in the plane of bending it will be safe to take about two-thirds of this, say, 26 in. tons as effective. The total maximum compressive stress is thus

$$\frac{13}{2 \times 2{\cdot}4} + \frac{26 \times 1{\cdot}66}{2 \times 6{\cdot}14}$$

$$= 2{\cdot}7 + 3{\cdot}52 = 6{\cdot}22 \text{ tons per sq. in.}$$

The allowable stress is either for $l/k_x = \frac{120}{1{\cdot}6} = 75$ for ends fixed or $l/k_y = \frac{90}{1{\cdot}24} = 72{\cdot}5$ for pin ends. The latter gives the lower value by $8 - {\cdot}04 \times 72{\cdot}5 = 5{\cdot}1$ tons per sq. in. This increased by 25 per cent gives an allowable compressive stress of 6·37 tons per sq. in.

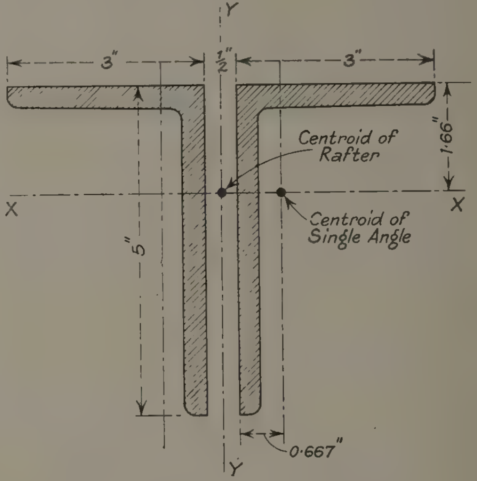


FIG. 46. SECTION OF RAFTER

As the minimum gyration radius of one angle is ·72, each angle must be restrained at intervals of $\cdot 77 \times 72\frac{1}{2} = 52$ in., so that a rivet through a $\frac{1}{2}$ in. washer at the centre of the rafter would be necessary.

If the purlin had been over a panel point a section consisting of two $3\frac{1}{2}$ in. × $2\frac{1}{2}$ in. × $\frac{5}{16}$ in. angles would have been ample. A smaller section having a gyration radius less than 1 in. would not be permissible, as l/k would then be greater than 120.

(CONCLUSION)

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